ENERGY USE AND ENERGY TECHNOLOGIES ON THE UNIVERSITY OF CANTERBURY CAMPUS

A thesis presented in fulfillment

of the requirements for the degree

of

Master of Engineering

in the

University of Canterbury

by

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1991

ABSTRACT

Building energy systems and the use of energy in an institution, (The University of Canterbury's Ilam campus) are investigated in this report. The existing installed systems are analysed and alternative "State of the Art" building energy systems are investigated. While technical and economic factors are the main criteria by which these systems are judged, commercial acceptability in New Zealand has also been a major concern.

Section 1 details existing campus building energy systems.

Section 2 examines the following alternative systems:

1. Peak Shaving of electrical demand peaks, is compared with energy cost savings from electrical load reductions.

2. Provision of both heat and electricity needs for the campus by Combined Heat and Power (CHP) plant. A CHP plant analysis spreadsheet was developed to help determine the performance of the plant under the varying simultaneous heat and electrical loads on the campus.

3. Alternative air conditioning systems are examined including, centralised district cooling schemes, evaporative cooling, and cold thermal storage.

4. Conversion of the existing steam heating system to hot water operation.

ACKNOWLEDGEMENTS

I would like to thank my superviser, Dr G J Parker, Senior Lecturer in the Department of Mechanical Engineering of the University of Canterbury. He has provided valued guidance throughout this study.

Professor A Williamson (Chemical and Process Engineering) and Dr A Tucker (Mechanical Engineering) have both provided helpful input in specific areas.

The staff of the Works and Services Department, Mr M White, R Lancaster, and D Lloyd, have freely provided essential information and helpful assistance during this study.

CONTENTS

LIST OF FIGURES

LIST OF TABLES

NOMENCLATURE

Effy	Efficiency	
Eisen	Isentropic Efficiency (of a turbine)	
EUF	Energy Utilisation Factor	
	EUF = Total Energy in / Total Energy out	
GJ	Giga Joule	
h	Specific Enthalpy	kJ/kg
hex	Spec. Enthalpy at turbine exhaust	"
hex(s)	Spec. Enthalpy at exhaust pressure with same spec. Entropy as inlet condition	"
hf	Spec. Enthalpy of liquid state	"
hg	Spec. Enthalpy of gaseous state	"
hin	Spec. Enthalpy at inlet	"
hout	Spec. Enthalpy at output	"
hout(s)	Spec. Enthalpy at entropy = entropy at inlet	
hpo	Spec. Enthalpy at pass out condition	"
hpo(s)	Spec. Enthalpy at pass out condition with same spec. Entropy as inlet condition.	"
kg	kilogram	
kJ	kilojoule	
kW	kilowatt	
kWh	kilowatt hour	
kV	kilovolt	
kVA	kilovoltamp	
LPHW	Low Pressure Hot Water	

MTHW	Medium Temperature Hot Water	
MW	Megawatt	
MWh	Megawatt hour	
Q	Heat	MW
Qbal	Heat balance (of a CHP plant)	MW
Qin	Heat input (of a CHP plant)	MW
Qout	Heat output (of a CHP plant)	MW
W	Power (Electrical load or output of CHP plant)	MW
Wbal	Power (electrical) balance of CHP plant	MW
Wout	Power output (of CHP plant)	MW
$M	Million Dollars NZ.	

INTRODUCTION

50% of the nett energy used in the UK economy is consumed in buildings, with industry using 29% and institutions and domestic dwellings the remaining 21%. [DUNSTAN I.]

While the UK climate, social and industrial situation is to an extent quite different to that of New Zealand's, the above values give an indication of the extent of energy use in the variety of buildings in most developed countries.

Commonly used energy conservation techniques popular since the 1970's oil crisis, based on loss reduction and load control, are generally applied throughout society and industry, giving significant energy and economic returns for little capital investment. However a stage is reached with conservative techniques where the performance limits of the (usually old) installed building plant fall well below the performance of more energy efficient state of the art building energy systems.

Industry and large institutions are of concern as the density of energy loads within these buildings are generally higher than those of dwellings.

In most cases it is difficult to remove existing building energy systems and install new state of the art systems, however existing plant can be modified, and new energy efficient systems can be installed in new buildings.

REFERENCES: All references listed by author or originator under section titled REFERENCES page 131.

1 EXISTING SYSTEMS.

1.1 ELECTRICAL SYSTEM.

1.1.1 CAMPUS ELECTRICAL SUPPLY AND DISTRIBUTION.

The University of Canterbury currently purchases its electrical supply from Southpower, the local electrical supply authority. Electrical supply is at 11kV, fed from both Fendalton Rd, and Ilam Rd substations,to an 11kV distribution ring main supplying eleven campus substations with 11kV/415V transformers, Figure 1.1 shows the campus high voltage electrical distribution system.

A 1987 report "Review of HT (11kV) Distribution" prepared for the university by the Ministry of Works and Development surveyed the installed electrical distribution system, its current performance, and capacity for future increase in demand. Table 1.1 below gives the installed capacity and present actual demand for the campus substation transformers.

TABLE 1.1. CAPACITY AND DEMAND OF CAMPUS SUBSTATIONS.

SUBSTATION	CAPACITY kVA	DEMAND kVA
1 Boiler house	200	85
2 Physics	1000	585
3 Geography	750	106
4 Students Union	300	170
5 Computer Centre	750	124
6 Library Arts	750	346
7 History	750	213
8 Engineering No 1	750	280
9 Engineering No 2	750	276
10 Geology	750	178
11 Forestry	750	177

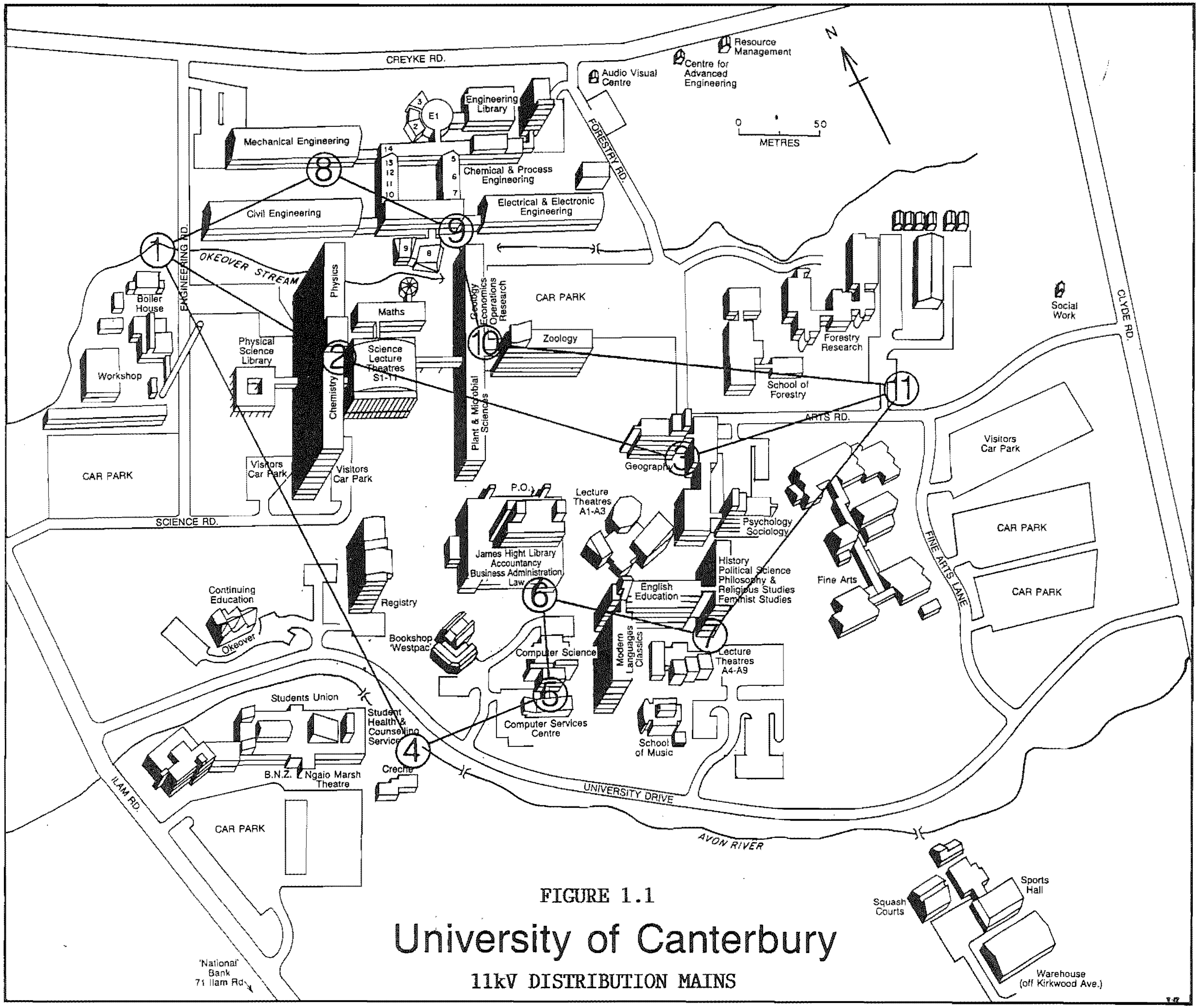


FIGURE 1.1

University of Canterbury

11kV DISTRIBUTION MAINS

It should be noted that while under the current state of reduced governmental support for the universities, little of the planned building program is likely to proceed in the near future. However all parts of the campus electrical distribution system are generously sized and in good order, and will easily accomodate significant development if required.

1.1.2 ELECTRICITY PRICING STRUCTURE.

Towards the end of this study the electrical supply tariffs offered by Southpower were altered with a greater emphasis put on supply capacity delivered to a consumer. Although this is still a function of peak demand, the emphasis on demand peak charges has been reduced. The following discussion covers the earlier supply tariff, with comparisons being made to the new tariff in section 1.1.4. New Tariff Structures.

The university purchased its electrical supply from Southpower under the ND6 coincident demand tariff, which was applicable to large non domestic users. Billing was on a monthly basis, and took into account factors such as, winter or summer rates, an energy charge, and charges based on electrical demand peaks, and demand peaks coincident with Southpower and Electricorp peaks, during the year up to the current month.

Figure 1.2 shows graphically the system of charges in the ND6 tariff.

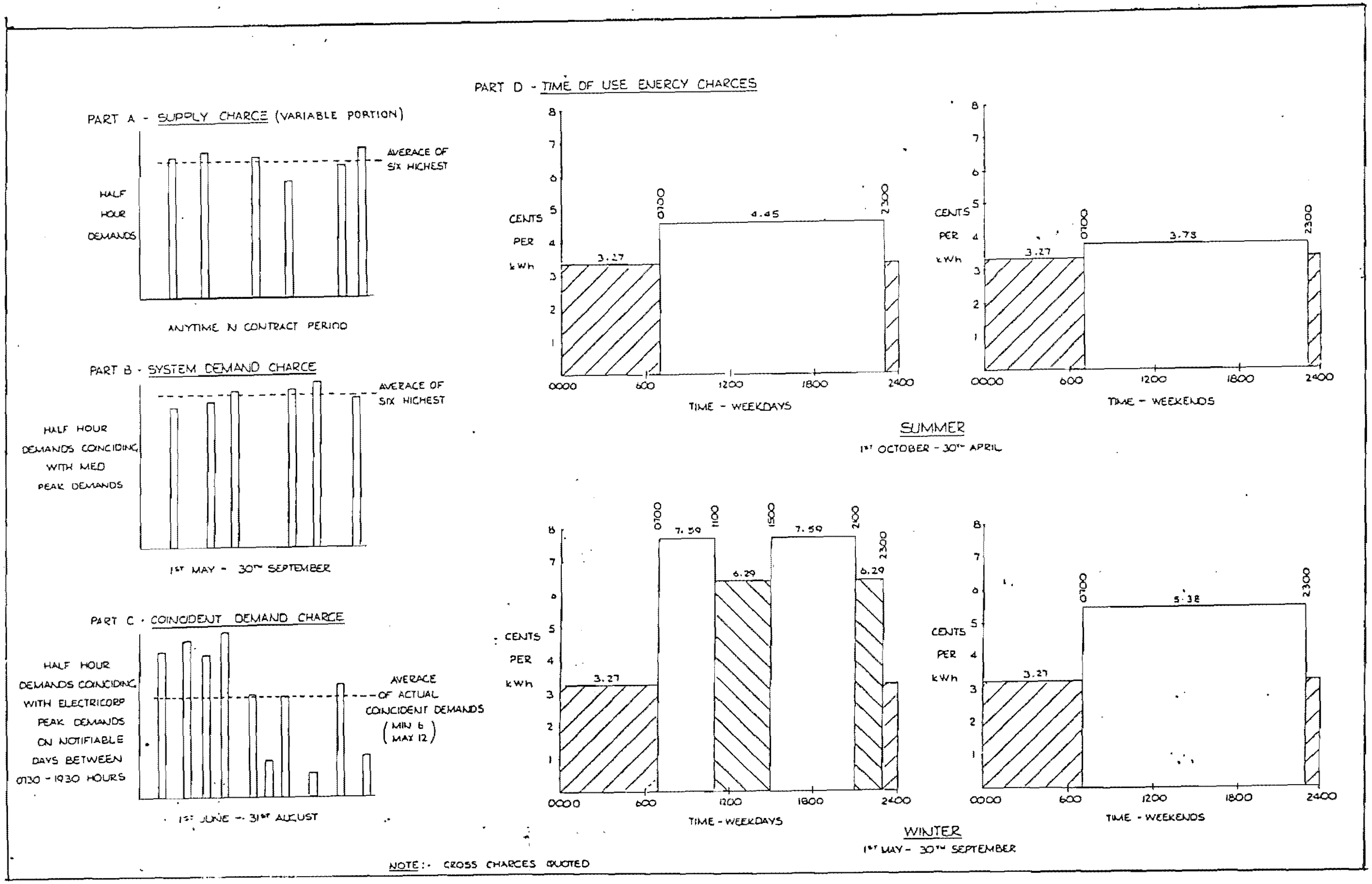


FIGURE 1.2 ND6 TARIFF

The ND6 tariff consisted of the following components;

A. Supply Charge.

1) Fixed portion - $6.90 per day per metering position.

2) Variable portion (11kV supply) - $28.80 per kVA of the average of the six highest half hour consumers demand peaks occuring anytime in the financial year to date. Only one demand peak per day will be charged. Any supply charge paid to date in the current financial year is subtracted from this charge, the remaining amount is divided by the number of billing periods remaining in the current financial year. This amount is the amount charged for that billing period

B. System Demand Charge;

$66.84 per kVA of the average of the six half hour consumers demand levels coinciding with the six highest demand peaks of the Southpower supply system in the five month winter billing period, (1st May to 31st September), with no more than one demand peak charged in any one day. Any system demand charge paid to date in the current financial year is subtracted from this charge, the remaining amount is divided by the number of billing periods remaining in the current financial year. This amount is the amount charged for that billing period

C. Coincident Demand Charge;

$87.60.per kVA of the average of the consumers half hour demand levels coinciding with the chargeable demands incurred by Southpower via the bulk supply tariff it has with Electricorp, between the hours of 7.00am and 11.00pm, on any weekday during the three winter months from 1st June to 31st

August inclusive. There will be a minimum of six and a maximum of twelve notifiable days in this period.

(Southpower was charged by the Electricity Corporation on the average of the six highest half hour demands incurred with the restriction of no more than one per day or two per working week.)

Any coincident demand charges paid to date in the current financial year were subtracted from this charge, the remaining amount was divided by the number of billing periods remaining in the current financial year. This amount was the amount charged for that billing period.

D. Fixed Energy Charge

1) Summer energy (1st October to 30th April)

7.00am to 11.00pm	Monday to Friday.	4.45c/kWh
7.00am to 11.00pm	All other days	3.78c/kWh
11.00pm to 7.00am	Night rate power	3.27c/kWh

2) Winter energy (1st May to 30th September)

7.00am to 11.00am	Monday to Friday	7.59c/kWh
11.00am to 3.00pm.	Monday to Friday	6.29c/kWh
3.00pm to 9.00pm	Monday to Friday	7.59c/kWh
9.00pm to 11.00pm	Monday to Friday	6.29c/kWh
7.00pm to 11.00am	All other days	5.38c/kWh
11.00pm to 7.00am	Night rate power	3.27c/kWh

The bulk supply tariff contract that Southpower had with Electricorp terminated on the 30th of September 1989. Changes in tariffs occured after that time.

A Total Load Indicator System (TLIS) was introduced by Southpower, to indicate to consumers, via a VHF signal the total load on the Southpower system. The Southpower peak load is about 450 MW

During the period 1988 - May1989 consumers were notified of coincident demand days on the day before the notifiable day. This was an experimental option which Electricorp withdrew from the supply authorities bulk supply tariffs.

Currently the half hour coincident peak demand periods are determined retrospectively by Electricorp, and charged on to the supply authorities.

Other Applicable Tariffs.

The only other suitable tariff for a large consumer such as the University was the ND4 Non Domestic Bulk Supply tariff. This tariff comprises the following components;

A. Energy charge;

6.27c/kWh at any time

B. Demand Charge;

$9.40 per kVA per month. Measured by a Maximum Demand Indicator which is reset each monthly billing period.

C. Supply Charge;

55c per day for each metered installation

A discount of up to 2.5% is available to consumers with their own transformers and 11kV metering.

In the Ministry of Works and Development report; "University of Canterbury Ilam Energy Investigations" the total charges for the University, using either ND4 or ND6 tariffs were compared for the following periods;

22 Nov 1985 to 31 May 1986 (Summer)

1 Apr 1986 to 23 Sept 1987 (Winter)

In both cases charges based on the ND6 tariff were lower.

With the ND6 tariff the consumer is charged for demand according to the part of the day / week / and year in which the power is consumed, allowing the supply authority (Southpower) to accurately pass on its increased costs of supply at its peak demand times. With the ND4 tariff the consumer faces flat energy and demand rates, which reflect the costs of an expected consumer demand profile, plus a cost associated with the risk of the consumer drawing heavy demands at times when the supply authority faces large demands.

For a consumer with an unavoidable constant demand at peak times, and little or no demand at off peak times, ND4 may be a more appropriate tariff than ND6, as that consumer will likely be unable to make use of significant amounts of cheaper off peak power. However if a consumer is able to maximise power consumption during off peak periods, then ND6 is likely to be a better tariff structure to operate within.

An important aspect of any comparison of the two tariffs is that in ND4 the consumers demand peak recorded for that month determines the demand charge for that month only, while under the ND6 tariff the consumers demand peak for any particular day may contribute to the demand charge for the next twelve months. Because the demand charge is based on the average of the six highest consumers demands in the financial year to date, an extraordinary demand peak (although it is averaged out with the five next highest demands) will increase demand charges by up to $183.24 per kVA of the increase in average demand.

It is also likely that the Southpower system peaks and coincident peaks will occur at the same time during the three month winter coincident peak charging period. Clearly for a consumer to maximise the benefits of supply on the ND6 tariff, the consumer must be capable of controlling demand during those periods when Southpower is sustaining chargable peaks.

Implications of ND6 Tariff.

The methods of charging employed in ND6 are of two types, energy charges,(Part A of the tariff), and demand charges,(Parts B,C, and D of the tariff).
Both energy charges and demand charges vary considerably for different periods, as the supply authority reflects in its charges the increasing marginal costs of extra generating capacity required at times of high power consumption. Southpower are simply charging a premium for power when the demand for that power is at a premium.

Energy charges may be reduced by the consumer by the following methods;

1. Minimise electrical consumption by using established energy saving techniques, such as automatic lighting control, solar shading, insulation standard improvement etc.

2. Maximising consumption during off peak periods, when lower charges are incurred. As little thought appears to have gone into off peak power usage during the development of the campus this will probably require some alteration to existing methods of energy utilisation on campus. Essentially any electrical load that doesn't require instantaneous consumption of electrical power, at the time of demand, has potential for off peak usage when combined with suitable storage facilities to make up the time lag between consumption and usage of the electrical energy.

Demand charges may be reduced by the consumer by the following methods;

1. Reduction of demand peaks, particularly at times of

Southpower system peaks and coincident demand periods. The reduction of a single kVA in coincident demand will save up to $183.00 per annum,(sum of the three demand charge components). This may be achieved by either, load shedding of any non essential electrical loads, and by rescheduling times of operation of some facilities and electrical equipment to avoid peaks in demand.

2. Peak shaving. Here demand over a certain level is satisfied by electricity generated by stand-by or base-load alternater sets, within the consumers complex.

Generally peak shaving involves running the generating sets only during the period in which system and coincident peaks occur. Without the notifiable days option which was withdrawn in May 1989, consumers have no way of knowing in advance of coincident demand days. This makes peak shaving an uncertain option as in order to be sure of reducing a coincident demand peak the consumer will have to guess the likely times these will occur, and run peak shaving generating plant for longer than necessary.

To an extent, any of the above activities will reduce both energy and demand charges simultaneously.

1.1.3 CAMPUS ELECTRICITY CONSUMPTION CHARACTERISTICS.

Daily Demand.

Typical daily power consumption is fairly steady, and daily demand profiles are similar to those of any other comparable day.

The campus daily electrical demand reaches maximum levels regularly at two distinct times;

11.30 to 11.45am, and 3.15 to 3.30pm.

Figure 1.3 shows a typical winter's weekday and weekend demand profiles, and a typical summer weekday and weekend demand profiles.

Southpower system and coincident peaks, have occured consistantly at two distinct times;

9.00 to9.30am, and 5.45 to 6.15pm.

These are highlighted on the daily demand profiles of Figure 1.3. Up to 1989 the university's demand at coincident and demand peaks were up to 30% lower than the campus daily demand peaks. A typical example is the Southpower supply demand peak drawn on the 16th of June 1988;

Campus daily peak demand	3.023 MVA @ 11.45am
Campus Demand at system peak	2.185 MVA @ 6.00pm

Currently Southpower system peaks are occuring at midday, coinciding with the campus daily demand peaks. This makes the need for demand control and reduction on campus greater.

There are no significant electrical loads in the university that could be shed during peak or coincident load periods, in order to effect a reduction in demand charges. Most of the electrical load is evenly distributed throughout the

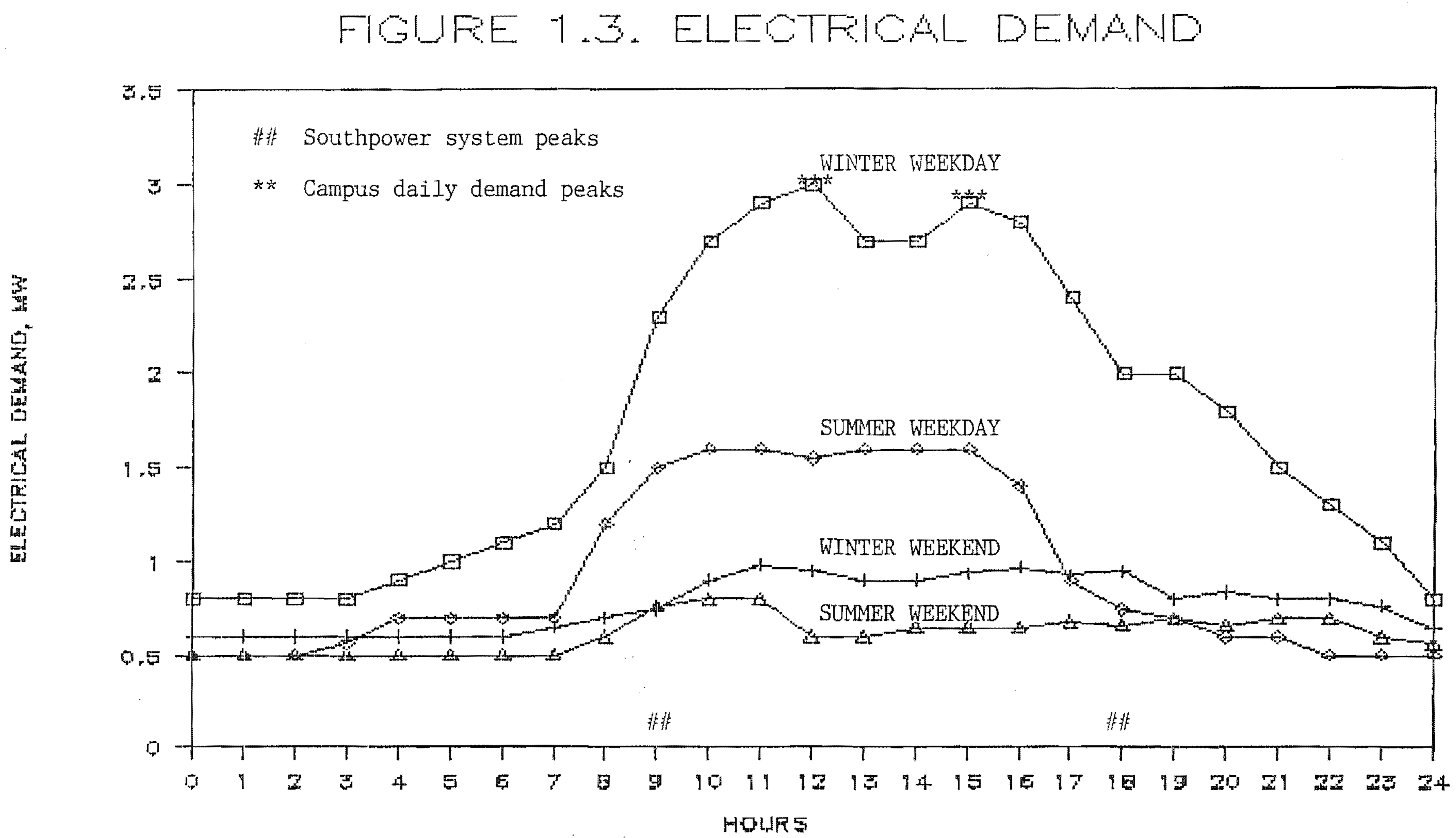


FIGURE 1.3 DAILY ELECTRICAL DEMAND PROFILES

campus, and required to be in operation during the day. Many parts of the university function through the evening till 11.00 pm. The libraries are in use until this time, as well as various post graduate studies, and cleaning staff are working at this time also.

Annual Electrical Consumption.

Figure 1.4 shows the monthly consumption of electricity by the university in 1988. Both power consumption in MWh, and total monthly charges are shown for each month. The graphs show electrical consumption increasing over the university term, despite the fact that heating is provided by the steam heating system, and air conditioning plant is not heavily loaded during this time of the year. Increased use of the university facilities during term time, (which coincides with the winter electricity charging period), and reduced daylight hours, would be expected to increase electrical consumption during term time.

The total amount of electricity consumed in 1988 was 9.010 GWh, at a total charge of $838,465. This gives an average cost per unit of 9.31c/kWh over the year. Power factor correction equipment was installed on the electrical distribution system in 1987-1988. Previously power factor was as low as 0.76 for most of the year. This is now improved to a value of about 0.96.

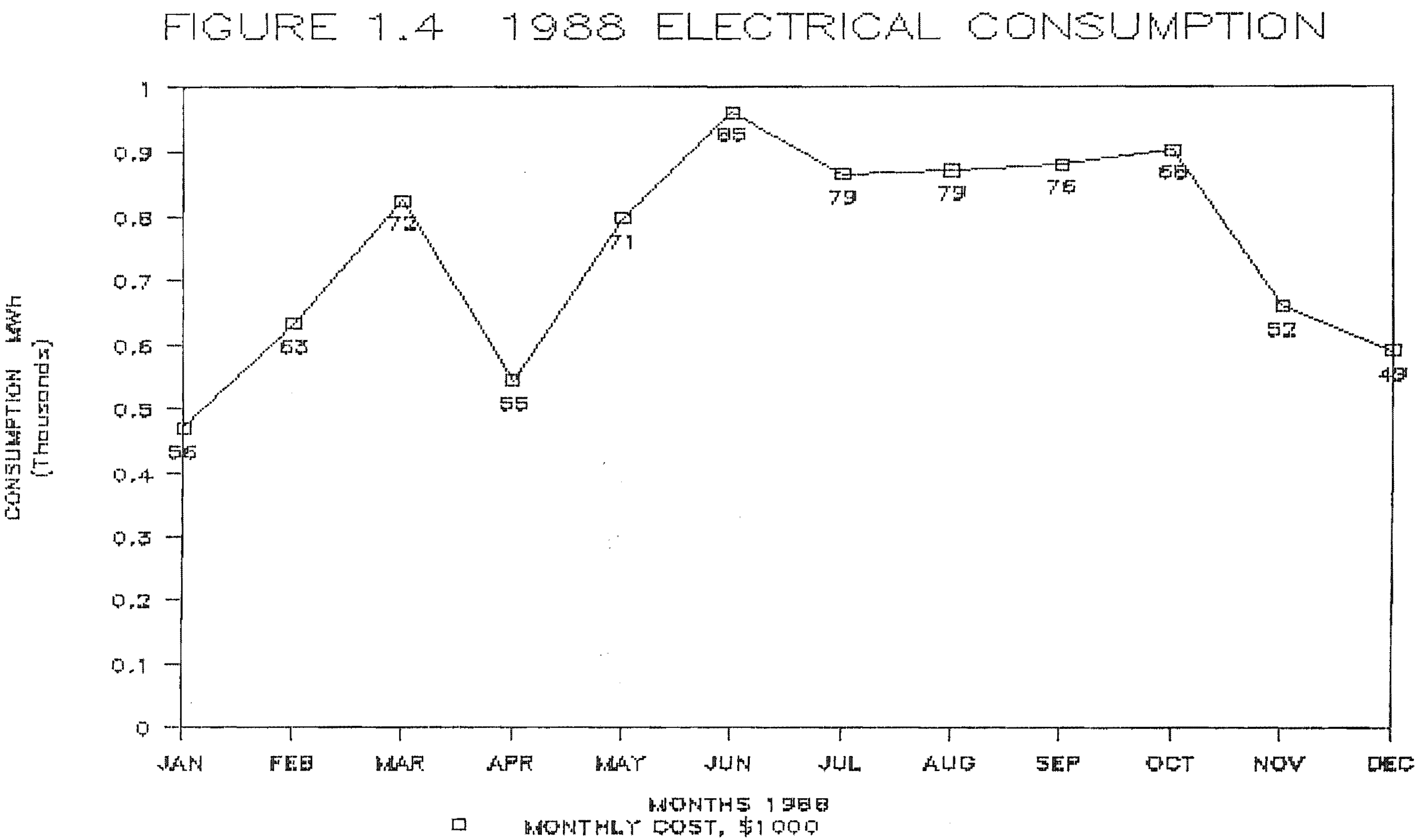


FIGURE 1.4 ANNUAL ELECTRICAL CONSUMPTION

1.1.4 New Tariff Structures.

Southpower introduced new tariff structures for all consumers in 1990.

The new tariff NM2, has a significantly reduced peak demand charge, however a new fixed capacity charge is included. NM2 includes the following charges;

A. Supply Charge

$7.06 per day per metering position

B. Capacity Charge

$13.32 per kVA capacity per day (for 11kV supply)

This is a fixed charge based on historical "registered demand" RD, and anticipated maximum demand "installed capacity" IC.

$$\text{Capacity} = RD + \{0.5(IC - RD)]$$

The registered demand is based on the average of the six highest half hour demands of the previous twelve month billing period.

C. Demand Charge

$27.74 per kVA per day for the average of the six metered half hour demands 7am to 11pm during June to August inclusive which are coincident with the six Southpower maximum demands in this period. No more than one peak per day or two per week.

D. Energy Charge

1) Summer energy (1st October to 30th April)

7.00am to 11.00pm	Monday to Friday.	4.61c/kWh
7.00am to 11.00pm	All other days	3.94c/kWh
11.00pm to 7.00am	Night rate power	3.38c/kWh

2) Winter energy (1st May to 30th September)

7.00am to 11.00am	Monday to Friday	7.88c/kWh
11.00am to 3.00pm.	Monday to Friday	6.53c/kWh
3.00pm to 9.00pm	Monday to Friday	7.88c/kWh
9.00pm to 11.00pm	Monday to Friday	6.53c/kWh
7.00pm to 11.00am	All other days	5.63c/kWh
11.00pm to 7.00am	Night rate power	3.38c/kWh

The capacity charge which is based on the consumers system demand still penalises consumers with a high peak demand to base load consumption, although the demand charge is not as extreme as in the previous tariff.

The capacity charge will always be greater than the registered demand as the installed capacity is always greater than actual peak demand. For the University the installed capacity is about 7.5 MW and demand peak about 2.9 MW, so the capacity charge will be about 5.2 MW. This reflects the supply authorities need to charge for its obligation to supply the maximum demand a consumers installation is rated for, rather than the peak demand the consumer is likely to draw. The consumer is required to pay for having oversized supply mains capacity, and because of this it would obviously benefit the consumer to reduce the installed capacity to as low a value as possible. This could be done by derating the main line fuses to lower values.

1.2. CAMPUS STEAM HEATING SYSTEM.

1.2.1 STEAM HEATING SYSTEM DESCRIPTION

The existing campus steam heating comprises of a central boilerhouse delivering dry saturated steam at 100 psi (6.89 bar) over the campus via a steam main distribution network in underground walk through ducts. Figure 1.5 shows the steam distribution network. Steam mains throughout the site are 150mm diameter, and run full diameter throughout the length of the underground ducts. The steam mains serving the School of Engineering are 100mm diameter.

Heat exchangers in each building provide low temperature hot water for building heating, and domestic hot water services. Condensate is returned to the boiler feed pumps by steam traps and condensate return pumps.

Steam is also supplied to the Department of Scientific and Industrial Research's, Ilam Research Centre in Creyke Road, and the Ministry of Forestry's, Forest Research Centre. Both these facilities are charged for steam consumed.
The University of Canterbury Student Union Recreation Centre, and the University Halls of Residence in Maidstone Road, are both suplied with medium temperature hot water by inground "Insapipe" pre insulated heating mains, via heat exchangers on the main steam heating system.

The central boilerhouse contains two Andersons 2000lb/hr (2.52kg/s) coal fired economic boilers. These were installed with the building of the science school in the early 1960's. The School of Engineering, which was opened first on the Ilam campus, was initially heated by a John Thompson Ltd 8000lb/hr

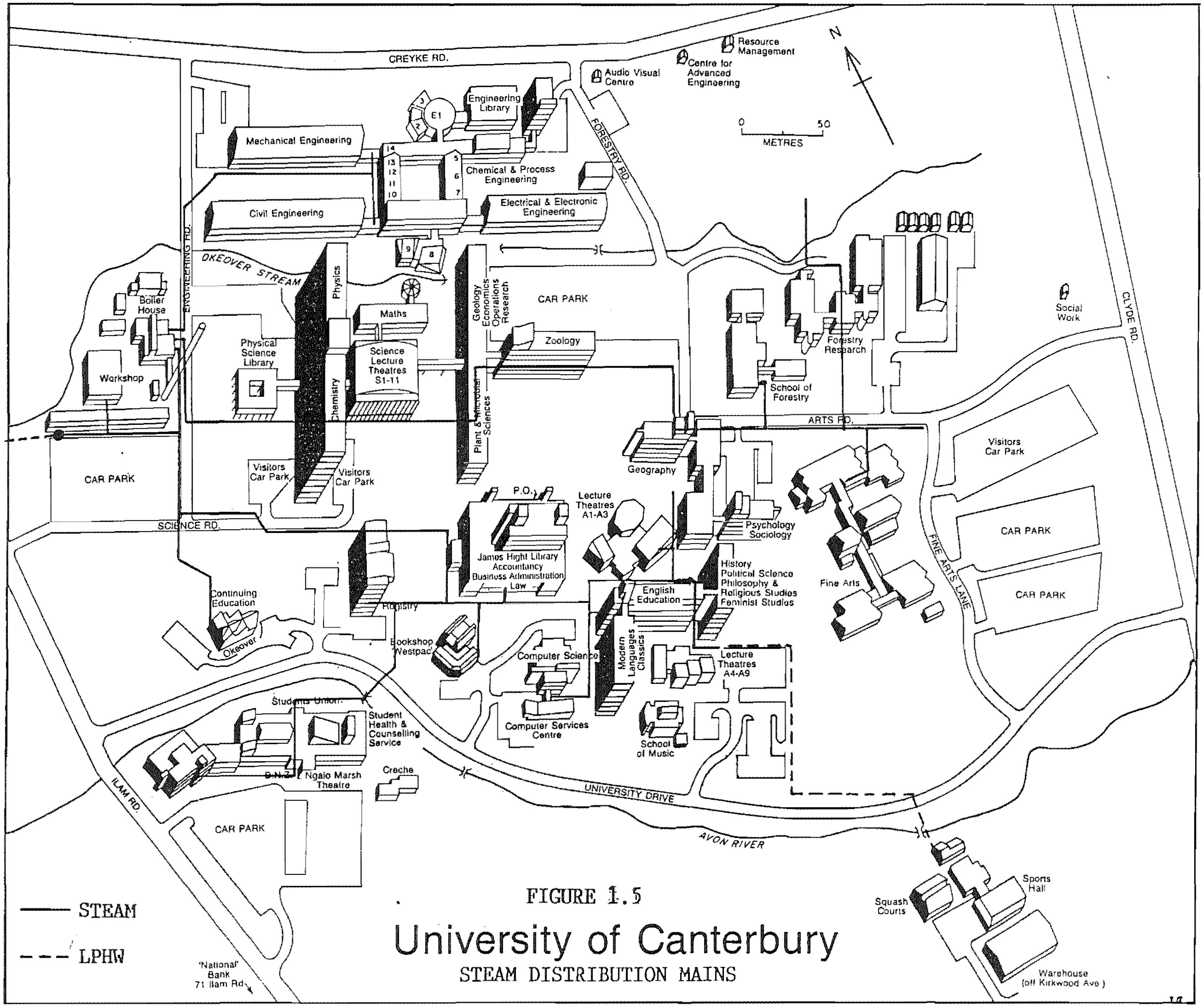


FIGURE 1.5

University of Canterbury

STEAM DISTRIBUTION MAINS

boiler. This is still in service alongside the two larger Andersons boilers.

Coal feed to the main boilers is by John Thompson chain grate stokers, and the main boiler feed pumps are Wier steam driven boiler feed pumps.

Heating System Performance.

All the main boiler plant is in excellent order, regularly well maintained, and has given reliable service. It is reasonable to assume that this reliability can be maintained for a considerable time.

While steam traps are notoriously problematic, and require continual maintenance attention, they are generally located in easily accesible locations, and are promptly attended to when requiring attention. The use of medium or high temperature hot water, for heating reticulation throughout campus would be desirable from maintenance, and energy control and conservation criteria, however the current record of reliable service of the steam distribution system would make the economics of converting the existing system from a maintenance point of view alone questionable.

While the steam distribution system appears to function satisfactorily, there are problems with the heating system performance within individual buildings. In some cases the heating system is unable to achieve the heating levels required in offices around the campus.

This is confirmed by the high level of electrical demand experienced on cold winter days, particularly in the mornings. A recent cold spell* set a new record peak electrical demand

* June 1990

of 3.400 MW. Previously peak demands had always been at about 3.000 MW, suggesting that there could be anything in the order of about 500 kW of personal electric resistance heaters scattered around campus.

The main reasons for the heating system problems appear to be due to changes in building usage over the almost thirty years of occupancy (eg Office space modifications occurring without complementary modification of the heating system). A common alteration appears to have been the conversion of large open plan offices into individual offices, with little consideration given to changes required to the control of the heating system. Anomalies in building shell construction and insulation play an important part in the current performance shortfalls of the system. Examples of this are ;

Until recently the top floor roof of the main library building was uninsulated.

The Botany / Geology building has a high mass building construction, but with a poor level of insulation, requiring a heating system start up time in the morning of about 3.00am, six hours before occupancy of the building.

The Engineering school laboratory wings are partly heated by uninsulated steam pipes running the length of the laboratories down the north and south walls, with virtually no control.

While the above examples are extreme, the experience of maintenance staff is that similar problems exist to a degree in most buildings on campus.

Generally at the time most of the buildings on campus were constructed, standards of building services control equipment were quite low. Current standards of quality of control are much higher, and upgrading of control equipment within buildings is being carried out by the maintenance department. Overall control of the heating system, and building services on campus is by a central computerised energy management and control system, (EMCS). The boilers are currently manually controlled by boiler attendants, but an automatic boiler control system is being installed, which will be supervised by the EMCS.

The increase in the level of comfort that is expected in buildings now, compared to the levels expected when the heating system was designed, only serves to make the above problems more conspicuous to the occupants.

It seems that the heating system components themself are adequately sized for the loads served, and that the boilers are providing sufficient steam to the heating system. Improvements to heating controls, building insulation, and the overall management of heating services by the energy management system, should be able to improve the performance of the heating system to a satisfactory standard, as well as reducing energy consumption.

Source;[Interview, D Lloyd Maintainance Dept. August 89]
[Unknown Member of Maintenance Dept., notes,1970'?]

1.2.2. STEAM PRODUCTION AND COAL CONSUMPTION.

Figure 1.6 shows the steam demand profiles for the following typical days.

Winter weekdays and Winter weekends

Summer weekdays and Summer weekends

The demands were developed from boiler steam flow recorders known to be inaccurate and as such give as good an indication of actual demands as possible. The winter weekday demand profile shows the maximum campus heating demand experienced on cold mid winter mornings.

The other demand profiles give an indication of the year round heating base load produced by domestic hot water demand and other miscellaneous heat loads.

The campus annual coal consumption shown on Figure 1.7 shows clearly the relationship between heat demand (and resultant coal consumption) and the seasonal variations in temperature. It should be noted that the January coal consumption value is lower than expected as the boilers are normally shut down for six weeks for maintenance.

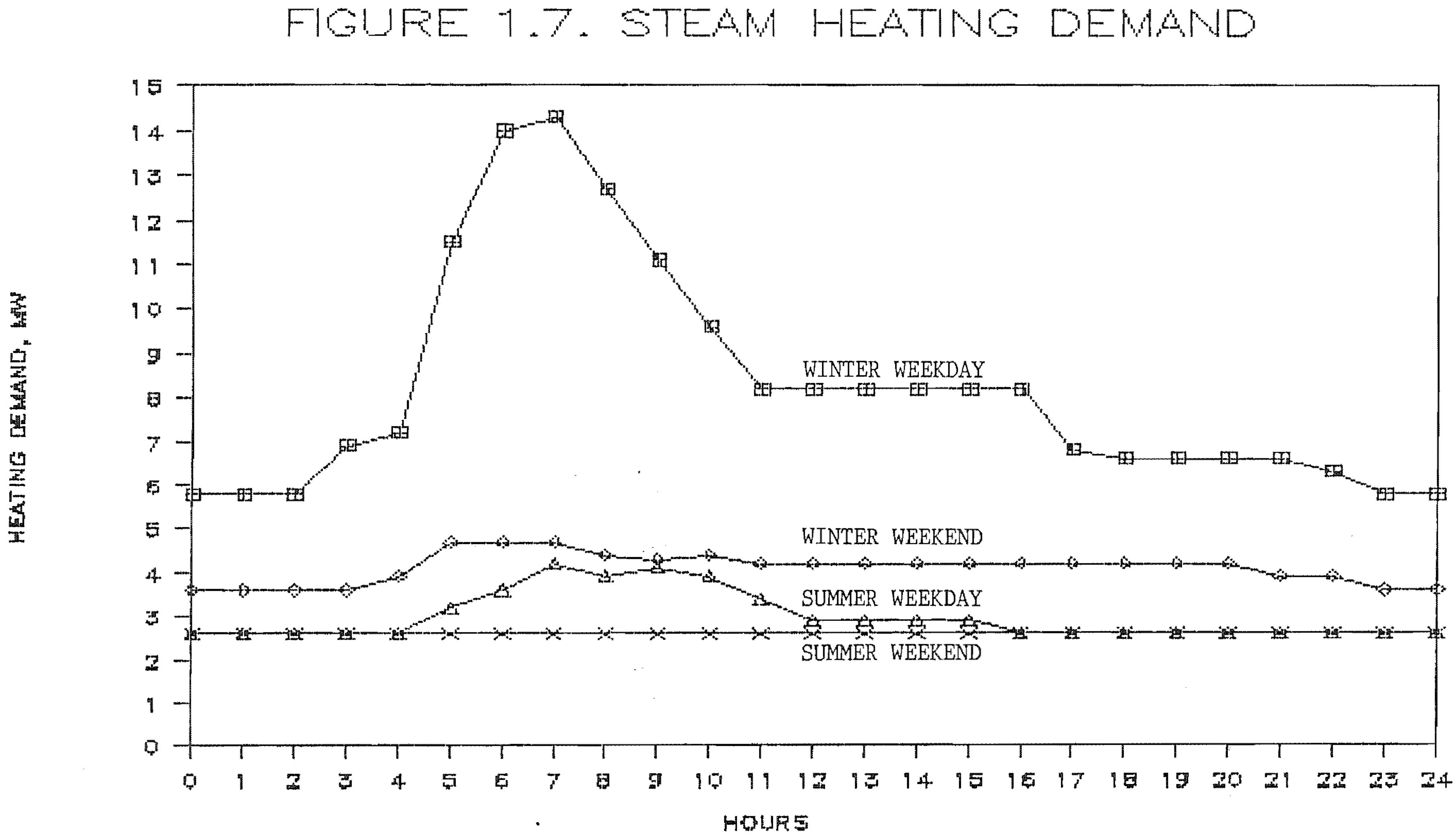


FIGURE 1.6 DAILY STEAM DEMAND PROFILES

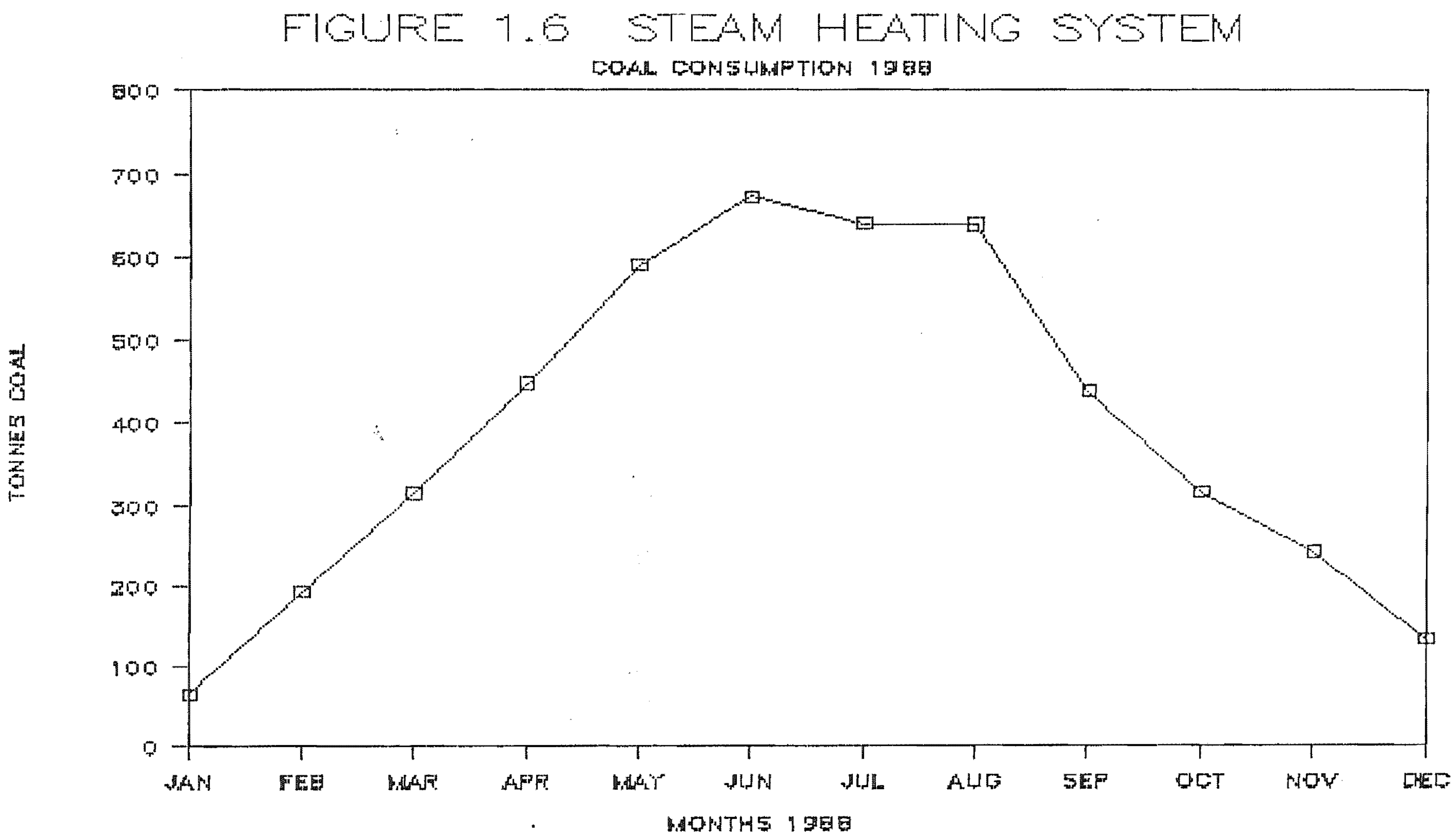


FIGURE 1.7 ANNUAL COAL CONSUMPTION

1.3 CAMPUS AIR CONDITIONING.

Cooling Load Distribution.

The university campus comprises many buildings spread over an area of 20 hectares. While there is a centralised heating system, the cooling requirements of the campus have always been met by the installation of individual "current state of the art" air conditioning plants when the buildings where constructed. The anticipation of a major building program in 1986 provided the initiative for an assesment of the state of the campus cooling systems. The Ministry of Works and Development (now Works Corporation of New Zealand) prepared a report titled "University of Canterbury Chilled Water Investigations."

The following section is a summary of the campus cooling systems based on the MOWD Chilled Water Investigations Cooling load Distribution

A breakdown of cooling requirements on campus is as follows;

Comfort Air Conditioning (Lecture theatres etc)	66%
Library Air Conditioning (Hight,Science,and Eng'g)	14%
Computer	8%
Specialist	12%

The largest users are the library and lecture theatres with a combined load of 80% of the total load. Computer cooling load is due to the computer centre. Remaining cooling loads are of small capacity and are spread out over many buildings.

Table 1.2 gives the actual capacity and type of cooling plant used.

TABLE 1.2 AIR CONDITIONING COOLING LOADS.

BUILDING	ARTESIAN WATER kW	DIRECT EXPANSION kW	CHILLED WATER kW
Registry		100	
James Hight Library	330		
Computer Center			134
Modern Languages		73	
English Education		12	
Arts N Lecture Theatres	317		
Arts S Lecture Theatres	92		
History			70
Psychology Staff		67	
Geog/Psych Lab Block		40	
Geography Staff		37	
School of Forestry		80	
Chemistry/Physics		70	
Science Lecture Theatres	460*		
Botany/Geology		13	
Zoology		10	
Engineering			
-Civil	48		
-Elec		24	
-Chem		1	
-Library		26	
-South Lecture theatres			170
-North Lecture theatres			224
-Middle Lecture theatres	202		
Totals	1499	553	598

The Science lecture theatres were cooled by an absorption chiller operating off the steam heating system, however operating problems present since its installation have meant that it has not been used in recent years. A lack of suitably skilled maintenance personnel in New Zealand for absorption machines is an important factor in this.

Direct expansion air conditioning plant are the most commmon type on campus. These are typically small units, less than 10 kW in capacity, which reject heat directly to the outside air. Often these are through the wall units which have been installed well after the building is in service, which have a high maintenance requirement, and are being installed in increasing numbers as the number of computers increase on campus.

66% of the total cooling load is due to occupancy, most of this from the lecture theatres, where cooling will be required all year round because of the occupancy density.
An important problem in an analysis of the university's cooling requirements is the lack of information about actual cooling demand patterns. Without a reasonable amount of basic data on the demands placed on the existing systems no accurate economic analysis of the benefits of various technical schemes is possible.

1.4 ENERGY MANAGEMENT AND CONTROL SYSTEM

An energy management system (EMS), is required to control building services plant operations in order to achieve a required level of environmental quality, while simultaneously minimising energy consumption and operating costs.

The EMS installed by the university's maintenance department is a Phoenix Energy Management and Control System (EMCS), produced by Staefa Control System Inc. This system ties up with the existing services control instrumentation without compromising the reliability of existing stand-alone control systems. If any failure on the part of the EMCS occurs, then the existing services control instruments revert to local, stand-alone operation. The EMCS operates in a supervisory manner that optimises the start up and shut down of the controls in each building on the campus, ensuring that buildings only come up to temperature when occupancy starts and are shut down before occupancy ceases.

The Phoenix EMCS system allows the use of Direct Digital Control (DDC) of systems with DDC remote multiplexing capability

System Architecture and Main Features.

Phoenix is a multi level distributed processing system. Each programmer supervises specific activities and performs specific functions. Each programmer typically reports to one higher level processor and supervises multiple lower level processors Each processing level is designed to report failures detected in lower levels and to continue operating without interruption in the event of a failure of a higher level.

The main features of the EMCS are;

1. Electrical Demand Limiting.

A power demand monitor gathers electrical demand data from the university and from the supply authorities system demand signals (Southpower provides a UHF signal for notifiable demand peaks) and produces a demand forecast.

2. Optimum Start and Stop Time Determination.

An optimum start time package provides a means of optimising start up times for the steam heating system for each building. The package evaluates information such as, heat build up rates for each building, environmental conditions, and estimates the start up time that will ensure the building is at the required temperature at the start of the occupancy period. Similarly the optimum stop time, allowing for the building's rate of cooling down, shuts of the heating systems before occupancy finishes so that the minimum amount of fuel is consumed.

3. Derivation of Data Values.

Important energy values that cannot be determined by a single sensor can be derived from data from several single sensors. An example may be the derivation of the value of enthalpy of fresh air, where the enthalpy may be determined from a dry bulb temperature sensor and a humidistat. Any number of sensors may be combined to give averages, maximum or minimum values, efficiencies, etc.

4. Centralised Time Clock.

The EMCS authorises all time clock functions on the campus from a central location. This allows out of hours scheduling of lecture theatres and start up / shut down of services plant to be easily controlled.

5. Energy Monitoring and Plant Monitoring.

Perhaps the most useful function of the EMCS is its ability to report and signal an alarm condition on plant operating status and system energy demand and consumption. This frees maintenance staff from time consuming plant checking duties, and enables instant feedback on any operational or system changes.

Specialised operational programs can be developed using the EMCS systems "Free Programming Language"

Several software features not associated with energy management are also available, eg security monitoring, fire alarming, planned maintenance scheduling, extensive data gathering and reporting, and colour graphics for building systems schematics. However as these are not essential energy management functions and are significant loads on both the computers memory and operating speed these are not used.

[Staefa Control Systems - Phoenix EMCS Prospectus]

SECTION 2. ENERGY TECHNOLOGIES FOR CAMPUS

2.1 ELECTRICITY USE REDUCTION OPTIONS.

2.1.1 ELECTRICAL PEAK SHAVING.

A significant part of the electricity supply cost was due to demand components of the ND6 tariff. A breakdown of annual electricity cost based on data for the 1988 year is as follows;

Supply Charge (Fixed Portion)	0.3%
Supply Charge (variable Portion)	6.6%
System Demand	22.8%
Coincident Demand	14.3%
Day Energy (various rates)	47.7%
Night Energy	8.6%

The combined effect of the demand charges amounted to 43.7% of the annual electricity account, while for the new NM2 tariff controllable demand charges amount to about 18% of annual charges. These charges are based on the electrical demand levels at specific times of the day. By generating electricity on site during coincident, and supply authority demand charge periods, the consumer may reduce the generally expensive demand component of its electrical account. This is achieved without using the large amounts of fuel required to generate the bulk of the daily electrical load.

An analysis of the possible savings due to peak shaving demand charges follows in this section. The main aims of this section are;

1. To determine if peak shaving is a viable option.
2. To establish an optimum level of peak shaving generating capacity, (if it exists).
3. To compare the effects of load reduction to those of peak shaving.

For this analysis the recoverable heat output of the generating plant is neglected, as this heat is of significantly lower thermodynamic and economic value than the electricity generated. Section 2.2, Combined Heat and Power Generation, examines further the heat and power requirements of the campus, and shows that extra heat production will be required in any case as the low grade reject heat from the generators will be insignificant when compared to the total campus heating load when the generation plant output matches the campus electrical load.

The following table shows the actual reductions in electrical charges resulting from peak shaving. Charges are based on the consumption data from electricity accounts for the period 21 July 1988 till 18 August 1988 and as such reflect the effects of the older ND6 tariff.

TABLE 2.1 ELECTRICAL PEAK SHAVING SAVINGS

Amount of peak shaved	0MW	1MW	2MW	3MW
Supply Charge 28 days	$193	$193	$193	$193
Supply Charge Demand	$7200	$7200	$7200	$7200
System Demand Charge	$12532	$8355	$4175	$0
Coincident Demand Charge	$16425	$10950	$5475	$0
Winter Energy.				
7.00 - 11 00 Weekday	$11150	$11150	$11150	$11150
11 00 - 15.00 Weekday	$11716	$11716	$11716	$11716
15.00 - 21.00 Weekday	$16665	$16665	$16665	$16665
21.00 -.23.00 Weekday	$2704	$2704	$2704	$2704
Weekend Day	$5884	$5884	$5884	$5884
Night Rate	$5442	$5442	$5442	$5442
Total Charges.	$89911	$80259	$70604	$60954

The calculated peak shaving charge reductions are the maximum that can be achieved for the installed peak shaving generating capacity. Fuel, capital, and maintenance costs have been neglected in this analysis as peak shaving generation involves operation only during the short demand chargeing periods. An indication of the operating costs for a half hour period are given later in this section. These range from $50 for a 500kVA generator to $290 for a 2900 kVA generator set. Generating periods are expected to be of about half hour duration, but where there is any uncertainty about the actual start and stop times of these, extended generation may be required in order to intercept the actual coincident or supply demand peak.

Where the coincident and system demand peaks do not coincide, (as occurs on campus) more frequent operation of the generating equipment will be required, resulting in a reduction of possible savings.

The low peak demand to average day time electrical load ratio of the university makes energy and supply charges a more significant part of the total electrical account, thereby reducing the suitability of the system for peak shaving. Peak shaving is ideally suited to consumers that have a short duration, high, demand peak that is coincident with the supply authority's peak demand charging periods.

Removal of the "notifiable days" option on the coincident demand part of the tariff, makes peak shaving difficult, as the consumer must now predict likely coincident demand chargeable days. While staff at the maintenance department have been able to predict the days in which coincident demand charges have occured with reasonable accuracy (4 days out of 6 in the 1989 winter period), by making a decision based on intuition and the weather forecast for that day, the operation of peak shaving generating equipment at incorrectly chosen times would soon eliminate the savings achieved.

Peak Shaving System Costs.

Because of the uncertainty of the frequency and duration of operation of a peak shaving plant, accurate assessment of operating costs becomes difficult. However a basic summary of the capital and operating costs of a range of peak shaving plants is given in the following table;

PEAK SHAVING PLANT COSTS.

Generator Capacity	Generator Installed Capital cost	Unit Cost, Half Hour Operation
500kW	$195,000	$50
800kW	$255,000	$80
1300kW	$420,000	$130
1600kW	$465,000	$160
2900kW	$885,000	$290

The above costs are based on the following;

1. Capital costs are for Diesel generator sets, Dual fuel or Gas engines are approximately twice as expensive.
2. Unit cost for half hour operation based on Diesel cost of $0.67 per litre giving a electricity production cost of $0.20 per kWh.
3. Installed cost of the generator set is taken as 1.5 times the capital cost. [SARGEANT W.]

2.1.2 ELECTRICAL LOAD REDUCTION.

The following table shows the actual reductions in electrical charges resulting from load reductions, achieved by either shedding or eliminating non essential loads, or generating on campus. Charges are based on the consumption data from electricity accounts for the period 21 July 1988 till 18 August 1988.

TABLE 2.2 ELECTRICAL LOAD REDUCTION SAVINGS

Amount of load reduced	0MW	1MW	2MW	3MW
Supply Charge 28 days	$193	$193	$193	0
Supply Charge Demand	$7200	$4800	$2400	0
System Demand Charge	$12532	$8355	$4175	0
Coincident Demand Charge	$16425	$10950	$5475	0
Winter Energy.				
7.00 - 11 00 Weekday	$11150	$5076	0	0
11 00 - 15.00 Weekday	$11716	$5027	0	0
15.00 - 21.00 Weekday	$16665	$7557	0	0
21.00 -.23.00 Weekday	$2704	$2704	$2704	0
Weekend Day	$5884	$5884	$5884	0
Night Rate	$5442	$5442	$5442	0
Total Charges.	$89911	$55988	$26273	NIL
Percentage Reduction	0	37%	71%	100%

Load reduction shows a considerably greater nett decrease in electrical charges than peak shaving, as the energy charges as well as demand charges are reduced. Changes to the methods of charging by electrical supply authorities are likely to occur in the future, in response to a trend for the electricity producer (or producers) to increasingly pass on the actual marginal costs of generation. As well as this the recent corporatization of the national electrical distribution grid means that consumers may be faced with a transmission charge component on their accounts. The long term variability of peak demand components of tariffs make peak shaving an economically dubious proposition, especially when similar savings could be expected from reductions in electrical load from improved electrical usage management. This is shown clearly by the reduction in the potential for peak shaving associated with the introduction of the NM2 tariff and its lower demand component.

The fact that significant reductions in electricity consumption could be achieved through the continued installation of modern high efficiency electrical fittings with optimum control, makes the installation of peak shaving generating equipment economically questionable. By making operational changes to the running of the university reductions in energy use can be made, however this may ultimately affect the effectiveness of the university's goals of teaching and research.

2.2. COMBINED HEAT AND POWER GENERATION.

Combined Heat and Power (CHP) production is widely recognised as an efficient method of simultaneously satisfying electrical and heating requirements for a variety of industries, public institutions, and residential schemes. While many different types of CHP scheme exist, the thermodynamic principle of minimising irreversible losses by utilising high grade heat for generating work, and low grade "reject" heat for useful heating, from a single heat source, is a common feature.

From an energy conservation point of view CHP provides a means of supplying energy services to society with a considerably lower drain on fossil fuel stocks. A 1989 report The Potential For Cogeneration in New Zealand [Ministry of Energy] concludes:

> "There is considerable potential for further cogeneration in New Zealand; as much as 300 GWh/year at a cost of 12c/kWh (the current Electricorp long run marginal cost of production) or less. The fuel efficiency for this level of cogenerated energy is expected to range from 65% to 80% ie between two and three times the fuel efficiency of Electricorp (thermal) generating plant."

Despite this apparently optimistic conclusion the level of CHP output in New Zealand is still quite low. The main cogenerators are the dairy and the pulp and paper industries. CHP production for institutions is at a very low level, and schemes that are in operation are marginal in their cost effectiveness. There would appear to be few technical problems associated with CHP plant. Both gas turbine and steam turbine (currently producing most of the cogenerated energy in New Zealand) alternator sets are available in a wide variety of sizes and configurations with good records of reliability.

Factors influencing CHP feasibility.

1. Energy costs. Both the relative price differences between different fuel sources and uncertainty over the future trends in fuel prices, affect the feasibility of various CHP schemes, and make reliable assesment of benefits difficult.

In New Zealand with a large supply of cheap hydro based electricity, the ability of any other generating system to produce electricity competitively is limited.

2. High capital costs of CHP plants make reasonable rates of return on investment hard to attain, and increase the level of financial risk to a CHP user, making other less expensive energy strategies more attractive.

3. Both heat and power load densities need to be quite high in order to keep energy distribution costs to a minimum. Proximity of the heat load to the power generating source is also important.

4. The ability to purchase standby electricity in the event of break down, without excessive financial penalty, and the ability to sell excess power at reasonable price, improve the feasibility of a CHP plant dealing with unmatched or varying loads.

5. Variable heat and power loads, especially where these occur independantly, make both design and operation of CHP plant uncertain.

2.2.1. CHP PLANT CHARACTERISTICS.

Steam Turbines.

This is the most common CHP plant in New Zealand, capable of being fired on coal, gas, wood, or waste fuels, high pressure steam (typically at 20 to 60 bar) passes through a turbine to a lower pressure steam process load.

The amount of energy required to raise steam does not vary much with steam pressure. The following table gives the energy required to raise one kg of steam from 1 kg of water at 20 deg C.

TABLE 2.3 ENERGY REQUIRED TO RAISE 1kg OF STEAM.

BOILER PRESSURE. (BAR)	SAT TEMP. (C)	ENERGY REQUIRED. (kJ)
1	100	2592
10	180	2695
20	212	2716
30	234	2720
40	250	2718

To raise steam at 40 bar requires very little extra energy input than that required to raise steam at 20 bar.

Boiler design limitations provide two main cost steps. Firetube shell boilers are inexpensive but limited to pressures of about 15 bar. At pressures above this more expensive water tube boilers are required until pressures of 45 bar and 450 deg C steam temperature require the use of special alloy steels, with disproportionately high costs for small installations.

For high turbine efficiency a large pressure ratio across the turbine is esssential, (necessitating as high as practicable inlet pressures) with some degree of superheat in order to limit blade erosion due to wet steam in the final turbine stages.

In CHP systems the turbine exhaust pressure is an important consideration. Unlike large power generating condensing turbines with turbine outlet conditions close to atmospheric temperature the heat or process load temperature requirements can considerably raise turbine outlet conditons. As a drop in exhaust pressure can give an increase in turbine output greater than that from an equal increase in turbine inlet pressure this is particularly important.

The following table gives turbine work output, and work to heat ratio as a function of exhaust pressure.

TABLE 2.4
EFFECT OF INPUT/EXHAUST PRESSURES ON TURBINE OUTPUT

TURBINE INLET PRESSURE (BAR)	TURBINE EXHAUST PRESSURE (BAR)	WORK OUTPUT (kJ/kg)	WORK / HEAT RATIO (Q/W)
40	10	315	0.13
40	7	378	0.16
40	2	588	0.20
40	1	684	0.32

See also Appendix A

The campus steam heating system operates with a boiler pressure of 100 psi (6.9 bar), which would form a severe limitation on turbine performance.

Turbine internal efficiency is also an important factor.

> "Although it is impossible to generalise due to the large number of permutations of steam pressures, temperatures and flows possible through a given turbine, a high speed geared machine is usually more efficient than a direct coupled machine for the smaller powers and exhaust volume flows. The gain in power more than offsets the gearing losses." [Valentine]

Most small turbines operate at about 7000 to 12000 rpm and require gearboxes.

Back Pressure and Pass Out Turbines.

Table 2.4 in the previous section gave the theoretical heat to work ratio for turbines operating with different exhaust pressures. The operating efficiency of a back pressure turbine can be optimised by selecting a back pressure such that the heat/work demand characteristic of the load is matched to the turbine. In order to use a back pressure turbine on the existing campus steam system a 7 bar exhaust pressure limit will be placed on the turbine as this is the pressure at which steam is currently produced for the campus heating system. This will produce a work/heat output ratio of 0.16 which is likely to be too low, as the typical work/heat ratio during daytime for the campus varies from 0.2 in summer weekends, to about 0.3 in winter. This means that at 7 bar exhaust pressure, supplementary boiler heating will be required throughout the year if the electrical load is matched. However if the campus heat load is matched by the CHP plant excess electricity will be available for sale. The work/heat load ratio of the campus varies considerably throughout the day as well as throughout the year, and at certain times of the year a better match between loads and turbine outputs will occur.

By lowering the exhaust pressure to 2 bar the turbine work/heat ratio will be close to 0.2 and will more closely match the campus work/heat load ratio. This will however put a maximum temperature limit of about 115 deg.C on the campus heating system. By converting the campus heating system to medium temperature hot water heating rather than steam this could be achieved, using a heat exchanger between the steam system and hot water heating system. This would still not optimise the turbine for any work/heat ratio other than the design one of 0.2. As the campus heat and electricity loads both vary considerably and independantly the efficiency of a back pressure turbine would be compromised frequently.

At this stage the desirability of using of a pass out or extraction turbine becomes apparent. Here steam is first passed through a high pressure (HP) turbine stage, to the heat or process load operating pressure, where it is either bled off to the heating load or further expanded through a low pressure (LP) condensing turbine stage. This significantly improves the turbine performance as the LP stage allows expansion of the non bled steam into the wet steam region down to pressures limited by the condensing medium's temperature. A pass out turbine for the campus would bleed steam at 7 bar to the heating system, however converting the heating system to MTHW operation would improve the turbine performance even further as the HP turbine stage would expand the full steam mass flow to a lower bleed pressure. The pass out turbine also allows much greater flexibility in matching the turbine heat and work outputs to the work/heat ratio of the load. Fig 2.3 shows the range of work/heat ratios that can be covered without importing or exporting heat or power for the campus.

While the need to match turbine characteristics to load is important, attempting to neither import or export heat or power may not be the overall most efficient use of the plant or fuel. As the electrical output is the most valuable, maximising this output (rather than losing potential work in incompletely expanded steam) and selling extra electricity is more efficient than operating at a lower efficiency for the sake of achieving a close match of loads to turbine output. For the university this would mean sizing the turbine to match peak heating loads and selling any extra electricity resulting. This shows that the ability to sell excess power without penalty is an important factor in the decision to install a CHP plant.

GAS TURBINES.

While steam turbines have traditionally been the main prime mover for thermal electricity generation, the rapid start up ability of gas turbines has made the gas turbine popular as a low capital cost peak load demand generator. In New Zealand, Electricorp has several gas turbine based peak load plants, ie Otahuhu, Stratford, and Whirinaki.

For CHP schemes the gas turbine is particularly suitable as the heat input not converted to shaft power is available as a high temperature exhaust gas (cf the low pressure low temperature exhaust steam from a steam turbine) which can readily produce a high grade process output. Compared to a diesel or dual fuel oil/gas engine where about 30% of the fuel input is rejected as low grade cooling water, this is quite significant. Both gas turbines and oil/gas engines produce an oxygen rich exhaust which can be further burnt with fuel to produce steam, this will however reduce the overall fuel utilisation factor as the extra fuel burnt in the boiler produces no work output.

Table 2.5 shows that generally gas turbine thermal efficiency falls between that of a condensing steam turbine and back pressure turbine.

TABLE 2.5 CHP PLANT TYPES AND THEIR EFFICIENCIES.

GENERATOR TYPE	WORK OUTPUT	USEFUL HEAT OUTPUT	ENERGY UTILISATION FACTOR
Gas Turbine	0.3	0.55	0.85
Condensing ST	0.4	0.0	0.4
Back Press ST	0.25	0.6	0.85
Pass Out ST	0.38	0.1	0.48
Gas/Oil Engine	0.36	0.3	0.66

[after Horlock]

However in CHP schemes thermal efficiency alone is not the most important factor in determining the suitability of a prime mover.

The ability of a CHP plant to maintain a high efficiency when following a wide range of varying electrical and heat loads is perhaps more important. This is especially important in the case of the university campus where heat requirements vary from 14.5 MW to virtually nothing and electrical loads vary from 2.9 MW to 0.5 MW.

> "The part load performance is also affected by the cycle chosen and by the method by which the (gas turbine) shafts are controlled. For instance a single shaft machine controlled to operate at constant speed for power generation tends to have poor part load performance, as at low speeds the compressor is forced to operate at full speed at part of its characteristic where it is not too efficient " [Craig]

An advantage of the gas turbine at part load operation is that if little or no heat load is required the turbine exhaust can bypass the heat recovery boiler to an exhaust stack, whereas a steam turbine always requires all the steam to be condensed before returning to the boiler feed pumps, requiring in some cases a considerable condenser and cooling tower.

Injection of high pressure steam from the gas turbines waste heat boiler into the turbine stage inlet, is a method which gives considerable flexibility to a gas turbine CHP plant. The Cheng cycle is an example of this, where steam injection mass is varied to enable a wide range of heat and electrical loads to be matched.

Koloseus and Sheperd describe the main advantages of the Cheng cycle as;

> "The added mass flow due to the steam produces an increase in work output from the turbine. For the Alison 501 the power output increases by 75% and the thermal efficiency by 40%. The price paid is 19% increase in fuel consumption."

Fig 2.1 shows the extensive operating range of a Cheng cycle series seven gas turbine.

As well, increased turbine life, and reduced NOx formation due to steam injection are claimed by Kelleher and Haselgrubler.

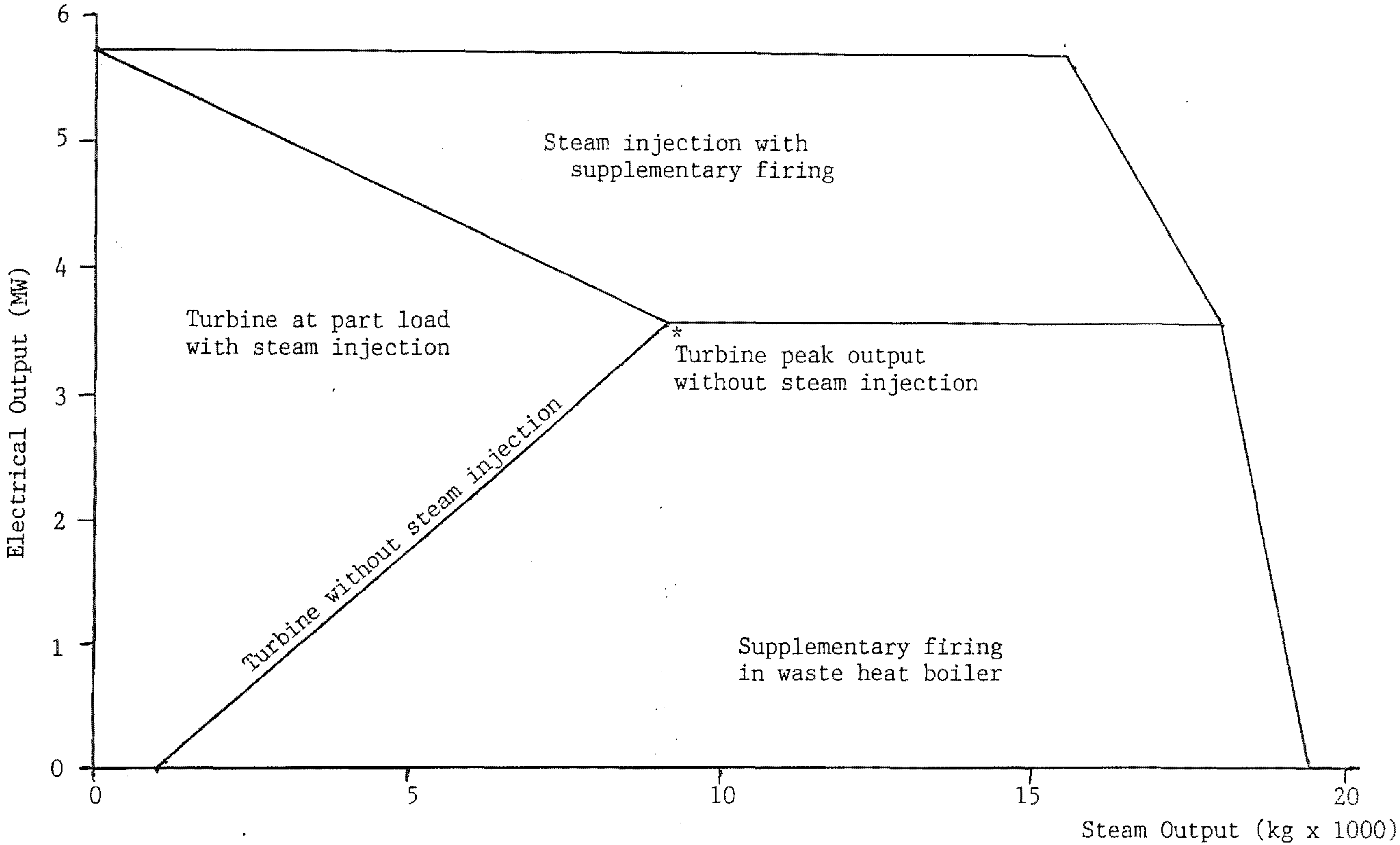


FIGURE 2.1 CHENG CYCLE OPERATING RANGE

2.2.2. Assessment of feasibility of CHP for Campus.

Design Heat and Power Loads.

An accurate estimate of the heat and electrical loads under which a CHP scheme will operate is essential to the sound analysis of the thermodynamic and economic performance of the CHP system.

Most industrial plants and institutions have complex, varying, heat and power load profiles, determined by the operating and environmental conditions. Methods for measuring the heat and power loads vary in complexity and accuracy, with more frequent measurements being required for continuously varying complex loads.

The simplest analysis involves taking a yearly average of heat and power loads and energy consumption, however this crude approximation is only suitable for the simplest initial feasibility study. Taking monthly averages of daily loads gives a more accurate assessment, reflecting diurnal and yearly load variations, resulting in 12 monthly average load profiles. If the actual hourly loads are determined from historical records the most accurate analysis of operating conditions can be built up, and a accurate assessment of actual operating costs and savings can be made. However the disadvantage of this is the large number of calculations required to produce this highly accurate result.

Bonham in a discussion of Ruston Gas Turbines TESOS (Total Energy Simulation Optimisation Study) computer based CHP plant simulator states:

> "-we have found by experience that a sufficiently accurate evaluation (of a CHP schemes performance) can

be made by considering only four typical operating days, namely

A typical winter production day,
A typical summer production day,
A typical winter non-production day,
A typical summer non-production day,"

[P BONHAM]

In the case of the university campus the above method of analysing the CHP loads is particularly appropriate, as the winter heating season almost coincides with the university term from March to October, and the typical daily loads are reasonably constant throughout the summer and winter periods. This method should give a sufficiently accurate assessment of the annual operation of the CHP plant, without requiring an excessive number of calculations.

A spreadsheet based load analysis program has been developed to enable a reasonably accurate estimate of the costs and benefits likely from the installation of a CHP plant on the campus. The following features are included:

1) The effects of variations in simultaneous daily heat and electrical loads are taken into account.

2) Required amounts of heat and/or electricity to be imported or exported resulting from imbalances between CHP outputs and load requirements are given.

3) Plant operation in two modes:

a. CHP plant satisfies electrical demand, heat may need balancing.

b. CHP plant satisfies heat demand, electrical load may need balancing.

4) Variations in plant thermal efficiency due to part load operation are taken into account.

5) Operating costs, including fuel and maintenance costs, and final heat and electricity output unit charges produced.

The CHP plant is assumed to operate continuously throughout the year, in order to maximise savings and return on capital cost, although in practice, operation in summer may not be economical.

Electrical tariff energy and demand charges are not included in the analysis for simplicity.

2.2.3. CHP SCHEME ANALYSIS RESULTS.

The following pages include summary sheets from the CHP analysis spreadsheets as well as details of the thermodynamic characteristics and economic factors used in the analysis. A complete example of one of the spreadsheets with explanatary notes is included in appendix D.

1. BACK PRESSURE TURBINE, 7BAR EXHAUST PRESSURE.

As no performance data relating to a specific production back pressure turbine was available, a performance model was developed based on an analysis of steam consumption of back pressure turbines by Kearton. See KEARTON 1964. The relationship between shaft power output, and total steam consumption is given based on the following design conditions:

Turbine steam inlet condition	40bar @ 400deg C
Turbine back pressure	7bar
Maximum power output	3MW
Turbine isentropic effy.	0.9

The relationship between inlet steam flow rate, and electrical load is shown on Figure 2.2 and is based on the

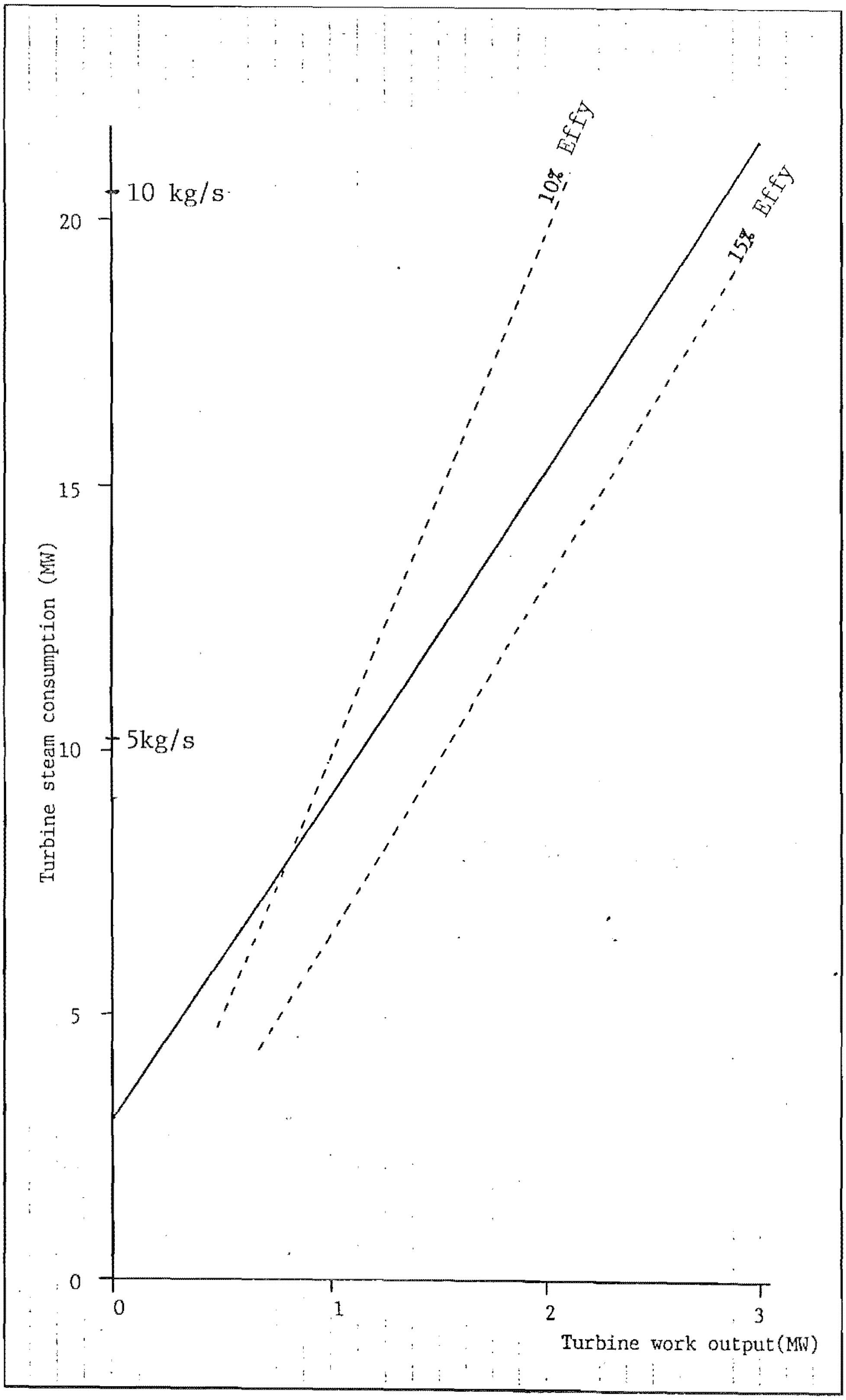


FIGURE 2.2 BACK PRESSURE TURBINE STEAM CONSUMPTION

calculations in Appendix A.

In realising this performance model on the CHP analysis spreadsheet, the back pressure turbine steam consumption is determined from the campus heat load, and simultaneous electrical load from the campus electrical load. The following relationship for turbine steam consumption is based on linear equations relating steam consumption to coincident heat and electrical loads.

$$Qin = 3.06MW + 6.28 * Wout\ MW$$

This is the function used to determine the thermodynamic performance of the back pressure turbine CHP plant when following the electrical load.

Similarly the analysis for the CHP plant following the heat load uses the linear heat load to steam consumption relationship, here the efficiency function is given by:

$$Qin = 1.189*Qout - 0.57MW/MW$$

For derivation refer Appendix A.

Economic analysis of the CHP plant is based on the results of the thermodynamic performance of the turbine, and the following economic factors:

Capital costs. (source Ministry of Energy 1989)			
Boiler (48 bar 450 c)			
	Cost		$M4.4
	Installation		$M1.0
Turbine / Alternator			
	Cost		$370,000
	Installation		$510,000
		Total	$M6.3
Maintenance costs.			
Boiler (5% capital cost / ann.)			$220,000
Turbine / alternator			$70,000
Fuel costs. (University of Canterbury)			
Coal ($88 / tonne @ 28MJ / kg LCV)			$11/MWh

Electricity (Ann. mean cost) $90/MWh

No attempt at detailed economic analysis is made, only a basic annual CHP plant operating cost is developed and compared to the present energy supply arrangement costs.

The following two pages sumarises the results of the CHP analysis spreadsheet

BACK PRESSURE TURBINE FOLLOWING ELECTRICAL LOAD.

PLANT NAME	BACK PRESS TURB	Th Effy function E = 9.4 + 1.5 *(EL	
FUEL: TYPE	COAL	Fuel Cost $/MWh	11.00
Rated work output	3 MW	Electy cost $/MWh	92.00
Heat cost $/MWh;	11.00	Maintenance $/ann	290000.00
Capital cost $;	6300000.00	Useful/rejectheat	1.00

PERFORMANCE RESULTS SUMMARY

Annual Steam MWh	34175.44	Stm.Charge.$/MWh	27.00
Annual Elect MWh	9471.28	Elec.Charge.$/MWh	92.00
Ann CHP Qin MWH	89515.64		
Ann CHP Qin cost	2567895.45		
CHP Qout MWh	61193.53	CHP Qbal MWH	-23444.73
CHP Qout cost $	1652225.19	CHP Qbal cost$	-670519.16
CHP Wout MWh	9471.28	Annual CHP cost $	3132376.30
CHP Wout cost $	871357.76	Current Ann.Cost	1251009.00
		Profit	-1881367.30
Mean Th Effy	10.58	Hours Operation at Rated Output.	3021.14
EUF cycle	0.79		
EUF system	0.7[illegible]		

BACK PRESSURE TURBINE FOLLOWS HEAT LOAD

PLANT NAME	BACK P. TURBINE	Th Effy function Qin=1.189*(Qout)-0	
FUEL: TYPE	COAL	Cost $/MWh	11.00
Rated work output	ELEC. 3MW	Elect'y cost $/MW	90.00
Heat cost $/MWh;	11.00	Maintenance $/ann	290136.00
Capital cost $;	6300000.00	Useful/waste heat	1.00

PERFORMANCE RESULTS SUMMARY.

Annual Steam MWh	37748.80	Stm.Charge.$/MWh	
Annual Elect MWh	9475.60	Elec.Charge.$/MWh	
CHP Qin MWH	39696.32		
CHP Qin cost $	436659.56		
CHP Wout MWh	1947.52	CH.· Wbal MWH	-7528.08
CHP Wout cost $	175277.09	CHP Wbal cost$	677526.91
Mean Th Effy	0.05	Hours operation at rated output	649.17
EUF plant	1.00	Annual CHP cost $	2349322.47
EUF system	1.00	Current ann. cost	1251009.00
		Profit	-1098313.47

2. PASS OUT TURBINE, 7BAR PASS OUT PRESSURE.

As no performance data relating to a specific production pass out turbine was available, a performance model was developed based on an analysis of steam consumption of passout turbines by Kearton. See KEARTON 1951. The relationship between shaft power output, pass out bleed steam, and total steam consumption is given based on the following design conditions:

Turbine steam inlet condition	40bar @ 400deg C
Pass out bleed pressure	7bar
Turbine exhaust pressure	0.1bar
Maximum power output	3MW
Maximum power - no pass out	1.6MW (Summer peak elec.)
Maximum pass out steam load	15MW
Turbine isentropic effy.	0.9

Two cases are developed, Full extraction, and No extraction of pass out steam in the analysis which is given in appendix B

The resulting inlet steam, pass out steam and electrical power relationship is shown on Figure 2.3. Although turbine design conditions will be different, the full extraction steam consumption line on Figure 2.3 shows the steam consumption of a back pressure turbine with exhaust pressure of 7 bar. Likewise the no extraction line shows the steam consumption of a condensing turbine with a condenser pressure of 0.1 bar. The ability of a pass out turbine to perfectly match any work / heat ratio between these two turbine operating extremes by controlling passout steam flow can be seen from this diagram.

Theoretical turbine electrical efficiency curves and pass out bleed steam rates are superimposed and show the improved work efficiency of the second stage condensing turbine, and the wide range of work / heat ratios the pass out turbine can accomodate.

In realising this performance model on the CHP analysis spreadsheet, the pass out bleed steam flow is determined from the campus heat load, and simultaneous electrical load from the campus electrical load. The following relationship for turbine steam consumption is based on linear equations relating steam consumption to passout steam flow and electrical load.

$$Qin = 1.12 + (Qout * 0.684) + (Wout * 2.433) \text{ MW}$$

This equation is the turbine efficiency function used in the CHP analysis spreadsheet to determine the thermodynamic performance of the pass out turbine CHP plant.

Economic analysis of the CHP plant is based on the results of the thermodynamic performance of the turbine, and the same economic factors as used for the back pressure turbine. In many cases a turbine manufacturer's back pressure turbines will be pass out turbines with blanked off pass out steam bleed points.

The following page sumarises the results of the CHP analysis spreadsheet

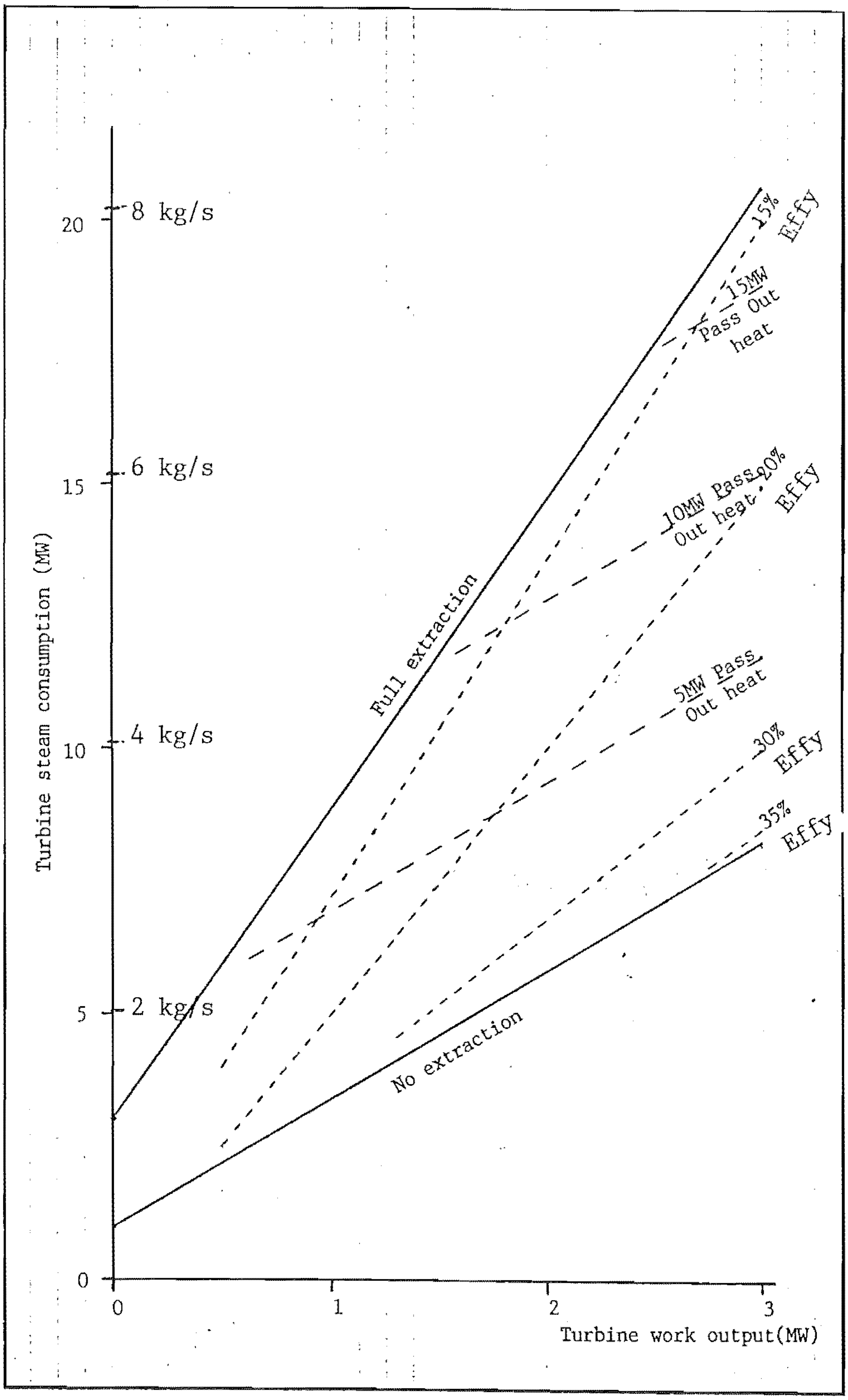


FIGURE 2.3 PASS OUT TURBINE STEAM CONSUMPTION

PASS OUT TURBINE - MATCHED LOADS.

PLANT NAME	PASS OUT TURBINE	Th Effy function Qin=(Qo*0.684)+1.1	
FUEL: TYPE	COAL	Cost $/MWh	11.00
Rated work output	Elec. 3MW	Elect'y cost $/MW	90.00
Qin cost $/MWh;	11.00	Maintenance $/ann	290000.00
Capital cost $	6300000.00	Useful/Qout Heat	0.95

PERFORMANCE RESULTS SUMMARY.

Annual Steam MWh	37748.80	Stm.Charge.$/MWh	
Annual Elect MWh	9475.60	Elec.Charge.$/MWh	
Ann CHP Qin MWH	59066.31		
Ann CHP Qin cost	649729.45		
Qcond	11841.91	Ann CHP cost $	1884729.45
CHP Cond. Capty.	2.83	Current Ann cost	1251009.00
		Profit	-633720.45
Mean Th Effy	0.15	Hours Operation	3158.53
EUFcycle	0.80		
EUFsystem	0.80		

3. GAS TURBINE, WASTE HEAT RECOVERY BOILER.

Although natural gas is not available in Canterbury in sufficient quantities to supply a CHP instalation, because of its importance as a CHP fuel, the relatively low capital cost, and high temperature heat output of gas turbines, a model was produced in order to give a comparison between gas and coal fired CHP systems.

For the Gas turbine analysis the performance characteristics of a production Alison 501k gas turbines are used.

The economic performance of the gas turbine is based on the thermodynamic performance and the following economic factors:

Gas Turbine Capital Costs.

Capital costs. (source Ministry of Energy 1989)

Gas Turbine (inc 3MW alternator)		
Cost		\$M2.3
Installation		\$510,000
Heat Recovery boiler		
Cost		\$M1.2
Installation		\$88,000
	Total	\$M4.1

Maintenance costs (annual) \$40400

Fuel Costs (source Wellington Gas Co)
(Monthly billing)
\$15.1/GJ for 1st 40GJ (\$54/MWh)
\$11.6/GJ for next 60GJ (\$42/MWh)
\$7.95/GJ for balance (\$28/MWh)
Mean Ann. Gas cost:

$$\frac{\$M1.269}{44998\ MWh} = \$28.7/MWh$$

(This assumes annual fuel consumption of 44998 MWh)

Electricity \$90/MWh

The thermal efficiency functions for the gas turbine are assumed to be linear, however in practise as the specific fuel consumption increases non linearly, and as power output drops the linear estimate will provide a more conservative value for turbine heat input.

The thermal efficiency function for the gas turbine following the electrical load is;

$$\text{Effy} = 14\% + 4.67\% * (\text{Wout MW})$$

For the case of the gas turbine following the heat load two gas turbines are assumed in the analysis, as one Alison 501k gas turbine alone is incapable of supplying the maximum campus heat load. The thermal efficiency function for the gas turbine following the heat load is;

$$\text{Effy} = 7.32\% + 1.19\% * (\text{Qout MW})$$

This assumes that the two gas turbines will operate in parallel although in practise they will probably be staged when the heat demands are sufficiently high to require both turbines to operate.

The analysis from which these functions were produced is detailed in Appendix C Gas Turbine Performance Calculations.

The following pages summarise the results of the gas turbine CHP analysis spreadsheets.

GAS TURBINE FOLLOWS HEAT LOAD

PLANT NAME	2x ALLISON 501-K	Th Effy function Effy=7.32+1.19*(Qo	
FUEL: TYPE	NAT'L GAS	Cost $/MWh	11.00
Rated work output	ELEC. 3MW	Elect'y cost $/MW	90.00
Heat cost $/MWh;	27.00	Maintenance $/ann	80800.00
Capital cost $;	5200000.00	Useful/waste heat	0.79

PERFORMANCE RESULTS SUMMARY.

Annual Steam MWh	37748.80	Stm.Charge.$/MWh	
Annual Elect MWh	9475.60	Elec.Charge.$/MWh	
CHP Qin MWH	46228.43		
CHP Qin cost $	508512.72		
CHP Wout MWh	6289.63	CHP Wbal MWH	-3185.97
CHP Wout cost $	566066.58	CHP Wbal cost$	286737.42
Mean Th Effy	0.14	Hours operation at rated output	1048.27
EUF plant	0.95	Annual CHP cost $	1656050.13
EUF system	0.96	Current ann. cost	1251009.00
		Profit	-405041.13

GAS TURBINE FOLLOWING ELECTRICAL LOAD.

PLANT NAME	ALLISON 501-K	Th Effy function	(ELEC LOAD)*4.67+1
FUEL: TYPE	NAT'L GAS	Fuel Cost $/MWh	28.75
Rated work output	3 MW	Electy cost $/MWh	92.00
Heat cost $/MWh;	27.00	Maintenance $/ann	40400.00
Capital cost $;	2600000.00	Useful/rejectheat	0.79

PERFORMANCE RESULTS SUMMARY

Annual Steam MWh	34175.44	Stm.Charge.$/MWh	27.00
Annual Elect MWh	9471.28	Elec.Charge.$/MWh	92.00
Ann CHP Qin MWH	49389.03		
Ann CHP Qin cost	1419471.83		
CHP Qout MWh	29654.01	CHP Qbal MWH	8094.79
CHP Qout cost $	800658.25	CHP Qbal cost$	231511.02
CHP Wout MWh	9471.28	Annual CHP cost $	2081382.85
CHP Wout cost $	871357.76	Current Ann.Cost	1251009.00
		Profit	-830373.85
Mean Th Effy	19.18	Hours Operation at Rated Output.	3021.14
EUF cycle	0.79		
EUF system	0.79		

Discussion of CHP Analysis Results

The performance of the CHP plants has been determined from theoretical plant operation models, and for the purposes of this feasibility study have been sufficiently accurate, however for improved accuracy, actual plant performance patterns should be used. Further accuracy would be achieved by using monthly load profiles rather than the four winter/summer daily load profiles. This would better show the effects on the plant of seasonal load variations. The load patterns for the steam system were taken from boiler steam flow charts known to have inaccuracies that may have overestimated the actual steam flow, however the totalled annual steam load equalled the actual annual steam production of the system.

Table 2.6 summarises the main cost / benefit factors of the five CHP systems ranked in order of annual savings due to CHP operation instead of the existing separate electrical and heat supply.

TABLE 2.6 CHP PLANT COSTS AND BENEFITS SUMMARY

CHP SYSTEM	ANNUAL SAVING	CAPITAL COST	OPERATING COST/ANN
GAS TURBINE HEAT LOAD	- $405,041	$M5.2	$M1.66
PASS OUT T. ELEC.& HEAT	- $633720	$M6.3	$M1.88
GAS TURBINE ELEC. LOAD	- $830373	$M2.6	$M2.08
BACK PRES.T. HEAT LOAD	- $1,098,313	$M6.3	$M2.35
BACK PRES.T. ELEC. LOAD	- $1,881,361	$M6.3	$3.13

The above results show clearly the important effect of high operating efficiency on CHP plant economic viability. Both back pressure turbine and gas turbine when following the campus electrical demand have low energy utilisation factors as a significant amount of the fuel is consumed at lower efficiency outside the CHP system to provide extra heat from conventional boilers.

The gas turbine following the campus heat load has the lowest operating loss, but less than half of its electrical output capacity will be consumed on campus, so any reduction in excess power purchase price from the local supply authority will futher reduce its viability.

While the pass out turbine has a perfect match to both the heat and electrical loads it is at the expense of overall efficiency. The system energy utilisation factor is almost the lowest of all the plants, due to the fact that some energy is lost in the condenser when the system is not operating in the full extraction condition. Compare this to the performance of the back pressure turbine following the heat load where all the turbine exhaust is condensed in the campus heating system, and at a resulting energy utilisation factor of one.

All the CHP plant configurations perform at considerably higher annual operating cost than the existing electrical and heat supply systems, and are therefore not viable alternatives. All the plants have high capital investment costs which cannot be recouped from the anticipated energy cost savings.

2.3 CAMPUS AIR CONDITIONING SYSTEMS.

2.3.1 THE REQUIREMENT FOR AIR CONDITIONING.

The university initially had a policy of only cooling large lecture theatres,and other areas where air conditioning was absolutely necessary.

Currently all lecture,tutorial,and teaching rooms as well as the libraries are air conditioned. This practise is the result of the present standard of thermal comfort expected by the public,and required by building services design standards. Previous building construction standards have tended to rather lightweight building shells with little thermal capacity to minimise the effects of temperature swings. As most of the buildings on campus were constructed about 20 years ago, insulation standards are lower than currently required. When this is combined with the low mass building construction, solar and occupancy gains require air conditioning in order to maintain design comfort conditions.

The main components of the air conditioning load on campus are;

1. Comfort Air Conditioning. Mainly lecture theatres and tutorial/teaching rooms.

2. Computer Cooling. The computer centre building and also an expanding demand for cooling for smaller computers distributed over campus.

3. Specialist Air Conditioning. Temperature controlled rooms and animal quarters in the Psychology department.

4. Library Air Conditioning.

Most of the large air conditioning plants on campus operate on economiser cycles where fresh air input to the system is modulated so that maximum use of free cooling from the fresh air or recirculated air is achieved. Depending on the weather conditions at the time a significant part of the design cooling load can be met by the free cooling from fresh air.

Currently extensive use is made of artesian bore water for direct cooling of air supply to conditioned spaces. The local authority (Canterbury Regional Council) estimates demand for artesian water consumption will exceed supply in twenty years time, and is attempting to control consumption. There is however no immediate threat to the university's artesian water supplies, as the Canterbury Regional Council can only alter a water right before its expiry date in the case of an extreme water shortage. The Council is currently pursuing a policy of requiring water users to meet the full costs of enforcing and monitoring water use, and part of the resource management costs. [DAMIANO 1990.]

It seems likely that in the future, water use charges will increase to reflect the actual marginal cost of supply of water resources, particularly as demand increases.

The current water rights are to expire in 1997, and it may follow that no new artesian water rights will be given for cooling water. The University sees itself as having an influence in the community, and is keen to set a responsible example for the use of the water resource. For this reason, as well as concern over the security of supplies, a policy of not installing any more artesian cooling plant has been developed. Because of this, research into alternative energy efficient cooling systems is necessary.

2.3.2 DISTRICT COOLING SCHEMES.

Where an appreciable air conditioning load exists in a densely concentrated area of buildings, a district cooling scheme supplying all the buildings cooling requirements from a central chiller plant can be an attractive alternative to individual chillers.

Chilled water at a flow temperature of 3 to 6 degrees celsius and a return temperature of 12 to 15 degres celsius is pumped to air conditioning plant in well insulated pipelines in a manner similar to the distribution of hot water or steam in district heating schemes. Worldwide, where there are climatic conditions requiring extensive air conditioning, district cooling schemes are employed in campuses, hospitals, industry, and built up areas of cities.

The central chiller plant may be either compression, or absorption types, and will often be associated with a district heating or cogeneration scheme.

The main benefits of centralised district cooling schemes are;

1. The capacity of the installed plant may be significantly less than the total cooling loads of the buildings supplied, due to diversity factors. This results in an overall decrease in operating costs as well as a reduction in capital expenditure.

2. Large chillers are more economical in operation and initial capital cost than a number of smaller chillers.

3. Low grade energy such as low pressure steam, or solar energy may be used for absorption chillers.

4. Reject heat from the chillers may be used to augment the heat supply of a district heating scheme, rather than rejecting it through cooling towers. Savings in the capital cost of cooling towers and energy savings from using this normally waste heat energy may be achieved. In the case of individual building chillers it may be difficult finding a suitable sink for the reject heat.

5. Centralisation of chilling plant.

> "Equipment is remote from campus activities, cooling tower noise and drift are localized, there are fewer operating personal," "efficiencies are higher at partial loads, and less space is required for machine rooms.
>
> [Wilson 1966]

6. Overall reliability of the air conditioning systems can be improved. Centralising the main plant allows for a degree of excess capacity, allowing standby operation, to be installed more cheaply than in the case for multiple system installation. Also for situations where temporary malfunctions occur the cooling capacity within a larger system can allow continued operation of essential services for a limited period.

> "It has been calculated that running costs for centralised plants can be as little as one third of those required for running individual air conditioning plants."
>
> [Diament R M E 1979]

As the built up area on campus is about 20 hectares the cooling load density is only about 13 W/m2. This is a value typical of air conditioned residential areas, and in this climate it is unlikely that a campus wide district cooling scheme would be viable. (as a comparison hospital complexes can have cooling load densities of up to 400 W/m2) There are however two areas on campus in which the cooling load density is considerably higher;

1. Engineering / Science lecture theatres.
2. Arts lecture theatres / Hight library.

Some degree of integration of chilled water systems in these areas could be feasible, especially as these builings require air conditioning throughout the year.

As with district heating schemes, the load density, ease of system installation, quality of scheme design, operating conditions and control, will all affect the economic and thermodynamic performance of a district cooling scheme.

Campus District Cooling Schemes.

The Ministry Of Works and Development report "University of Canterbury Chilled Water Investigation." [Ministry of Works and Development 1987] described three centralised chilled water options as follows;

1. A single central chiller complex with chilled water distributed via underground mains.

2. Two chiller complexes each one distributing chilled water to a localised group of buildings.

3. Provide each building with its own chiller plant, sized accurately to handle the total building load, as opposed to installing a multiplicity of small DX (Direct Expansion

)units.

Because of the low cooling load density option 1 would not be economically viable.The cost of distribution mains and pumping power would be prohibitive.

Option 2, the two chiller complex option avoids the high distribution costs associated with option 1 by situating the two chiller plant complexes close to the major cooling loads. The report proposes that the chiller complexes are situated as follows;

1. An "Arts" chiller plantroom incorporated in the commercial center development proposed at the time the report was proposed.

2. An Engineering/Science chiller complex in a stand alone building, possibly underground. For both systems cooling towers would be situated on the rooftops of adjacent buildings and chilled water mains pipes run in existing underground walk through ducts where possible. At the connection points in buildings the incoming chilled water mains will be connected to the existing air conditioning systems which will require modification depending on the type of existing equipment.

In option 2 all the advantages of centralised chilled water systems can be realised without the excessive costs due to the long distribution mains required in option 1.

The third option is a refinement of what has been done up until now. Individual chillers would supply chilled water to individual buildings, or in some cases to more than one building, with heat being rejected either by cooling tower or air cooled condenser depending on the capacity and type of chiller installed. Chilled water would be distributed around buildings to specific locations requiring cooling. Existing artesian water and DX systems would be converted to chilled water. The benefits of a centralised chilled water system would be lost with this system, but the option of connecting up the seperate chilled water systems in the future exists.

Costs for systems 2 and 3 as outlined in the Campus Chilled Water report are as follows;

Option 2; 2 Central Chillers.

2 Chiller Plantrooms	$500,000
Chillers	$720,000
Pumps and equipment	$300,000
Reticulation Mains	$200,000
Service Ducts	$1,100,000
Cooliing Towers	$150,000
Electrical	$200,000
Controls	$150,000
	$3,320,000
Design Fees & Contingencies	$680,000
Total	$4,000,000

The above costs include all costs associated with the connection of buildings included in the 1986 University capital works programme as well as connection of all existing buildings. As the building programme virtually doubled the air conditioning load on campus, and the cost of new walk through service ducts makes up one third of the cost of option 2, as this is largely due to the connection of new buildings over which there is some doubt if they are to be built, the costs of the scheme as listed are obviously not applicable to the use of an option 2 centralised chilled water system for the existing campus.

A revision of the above costs to derive a rough order of costs for a central chilled water system for the campus is based on the following assumptions;

1. Only large loads in close proximity to the central chiller plant should be connected to the system. The costs of connecting a small chiller in a remote building will far outweigh any possible savings, especially when the electrical loading of a compression air conditioner is only about 30 to 40% of its chilling or rated capacity as it has a COP of about 2.5 to 3. By applying this criteria to the systems, uneconomic conections will be avoided and the benefits of centralised chilled water systems applied to plant which will show the maximum savings.

2. With the reduction in chiller capacity due to the building programme not proceeding the construction of new underground chiller plantrooms and new underground ducts can probably be avoided by incorporating the new chiller plant in existing service spaces.

Rough order of Cost of two central chilled water systems based at Engineering/Science lecture theatres and Arts lecture theatres.

Chiller Plantrooms alterations	$50,000
Chillers	$410,000
Pumps and Equipment	$200,000
Reticulation Mains	$100,000
Cooling Towers	$90,000
Electrical	$200,000
Controls	$150,000
	$1,200,000
Design Fees & Contingencies	$245,000
Total	$1,450,000

OPTION 3

From the MOWD report the conversion of all the existing artesian water, and Direct Expansion plant to individual chilled water operation was estimated to cost $850,000.
All the above figures are based on 1987 equipment and construction costs.

Operating Costs

Electrical consumption and maintenance charges are the main operating costs associated with air conditioning plant. While both costs should be significantly lower for the central chiller plant system (Option 2) than for the individual chiller installations (Option 3), the actual costs are difficult to assess without reasonably accurate air conditioning demand data. The MOWD report estimated a saving in operating costs of $11000 annually for the central chiller plants, when compared to the individual installations, however this was for the case of satisfying the additional load due to the building expansion programme. Very little is known about the actual operation of air conditioning plant on campus, and even rough estimates of costs are difficult to make.

For any centralised air conditioning plant for this campus the capital costs are massive and are unlikely to be recovered by any possible energy savings. The low cooling load density and the fact that the cooling demand exists only for short intermittent periods in summer are the main reasons for the limited economic viability of any centralised scheme.

2.3.3. EVAPORATIVE COOLING.

Evaporative cooling is an energy efficient, cost effective method of cooling suitable for comfort air conditioning in institutional buildings in dry hot climates.

In investigating the suitability of the Christchurch climate for evaporative cooling a weather data file was used to determine the actual extremes in dry bulb and wet bulb temperatures. The file (Climdata 69 generated by the NZ Meteorological Service) contained hourly dry bulb and relative humidity data for the year 1969. Coincident dry bulb and wet bulb data were derived from this and analysed to determine the extremes limiting an evaporative coolers performance.

Evaporative Cooling Of Air.

1. Direct Evaporative cooling.

Evaporative coolers, air washers, sprayed cooling coils, or humidifiers are used to cool air directly by evaporating water directly in the incoming air stream. The dry bulb temperature is lowered as the air condition moves adiabatically up the wet bulb line corresponding to the air's wet bulb temperature. As the moisture content increases with direct evaporative cooling the resulting conditioned air has a higher humidity.

2. Indirect Evaporative Cooling.

A secondary air stream is cooled by direct evaporative cooling, this sensibly cools the fresh air stream to the conditioned space through a heat exchanger. Figure 2.4 shows the two processes on a psychrometric diagram.

Because the final dry bulb temperature is determined by the wet bulb temperature of the incoming fresh air, the performance of an evaporative cooler is directly related to the climatic conditions. Evaporative cooling is therefore best suited to dry cool climates with low wet bulb temperatures.

Basic weather data for the Christchurch area suggest evaporative cooling of air for air conditioning may be quite successful:

1% Std. Dev.DB Temperature	27.8 deg C
Mean Coincident WB Temperature	17.8 deg C
2% Std. Dev.DB Temperature	26.0 deg C
Mean Coincident WB Temperature	17.2 deg C

[Watt 1986]

Figure 2.5 shows the ASHRAE comfort standard 55-74 plotted on a psychrometric chart, with the above maximum DB, and coincident WB temperatures plotted. Also a typical summer peak hot day (Jan 19 1969 from Climate data file) hourly temperature profile is plotted. It can be seen that even allowing for less than ideal cooler performance, direct evaporative cooling should be capable of achieving the necessary comfort conditions on this worst case day.

The actual size and type of evaporative coolers used will depend on the total load of the air conditioned space, and the use of that space. The libraries for instance may require closer control of humidity than that available from an evaporative cooler, and require the use of indirect evaporative cooling.

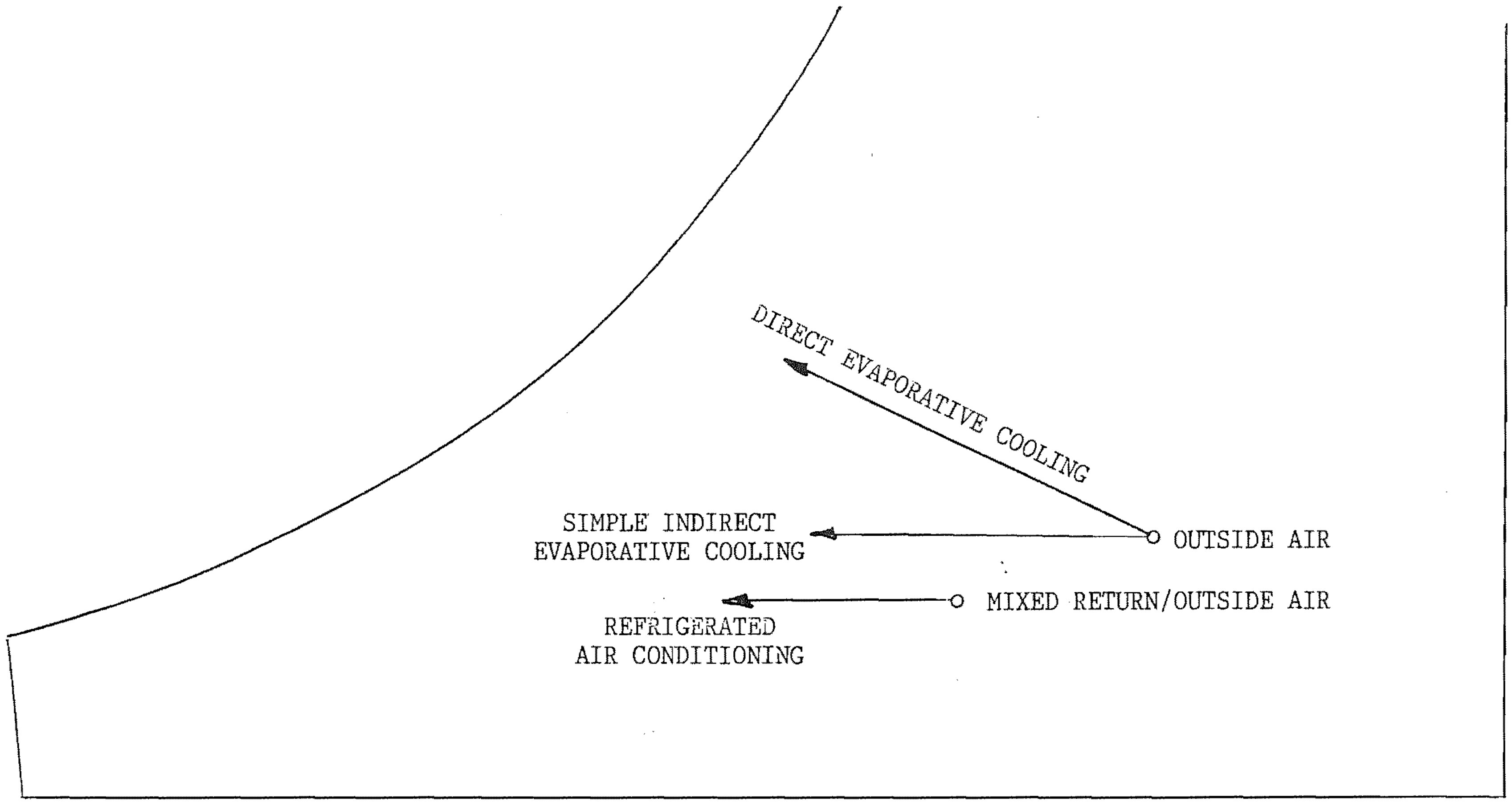


FIGURE 2.4 DIRECT & INDIRECT EVAPORATIVE COOLING

Evaporative Cooling Of Chilled Water.

The existing artesian water supply to cooling coils on campus is at a temperature of about 12 deg C. The possibility of achieving this temperature for air conditioning chilled water by evaporative cooling is addressed in this section. The evaporative cooling of a liquid is both a heat and mass transfer process occurring on the free surface of the liquid by the combined effects of:

a)heat transfer by conduction

b)heat transfer by radiation

c)surface evaporation of the liquid

Cooling water is recirculated through an air flow,cooled by evaporation, while the enthalpy of the air increases as it becomes more humid.

Make up water for evaporative losses is of the order of 1 percent of the circulation rate [Jones 1989]
As with evaporative cooling of air the wet bulb temperature, and particularly its depression below dry bulb temperature determine the applicability of evaporative cooling.

At night, wet bulb temperatures are lower than during the day, (see Figure 2.5) and evaporative cooling efficiencies are improved. In a climate where evaporative cooling is unsuitable for use during the day the application of chilled water storage combined with evaporative cooling of the chilled water to charge up the storage volume at night can make evaporative cooling attractive. An added benefit is that the cooling plant is run on night rate electricity.

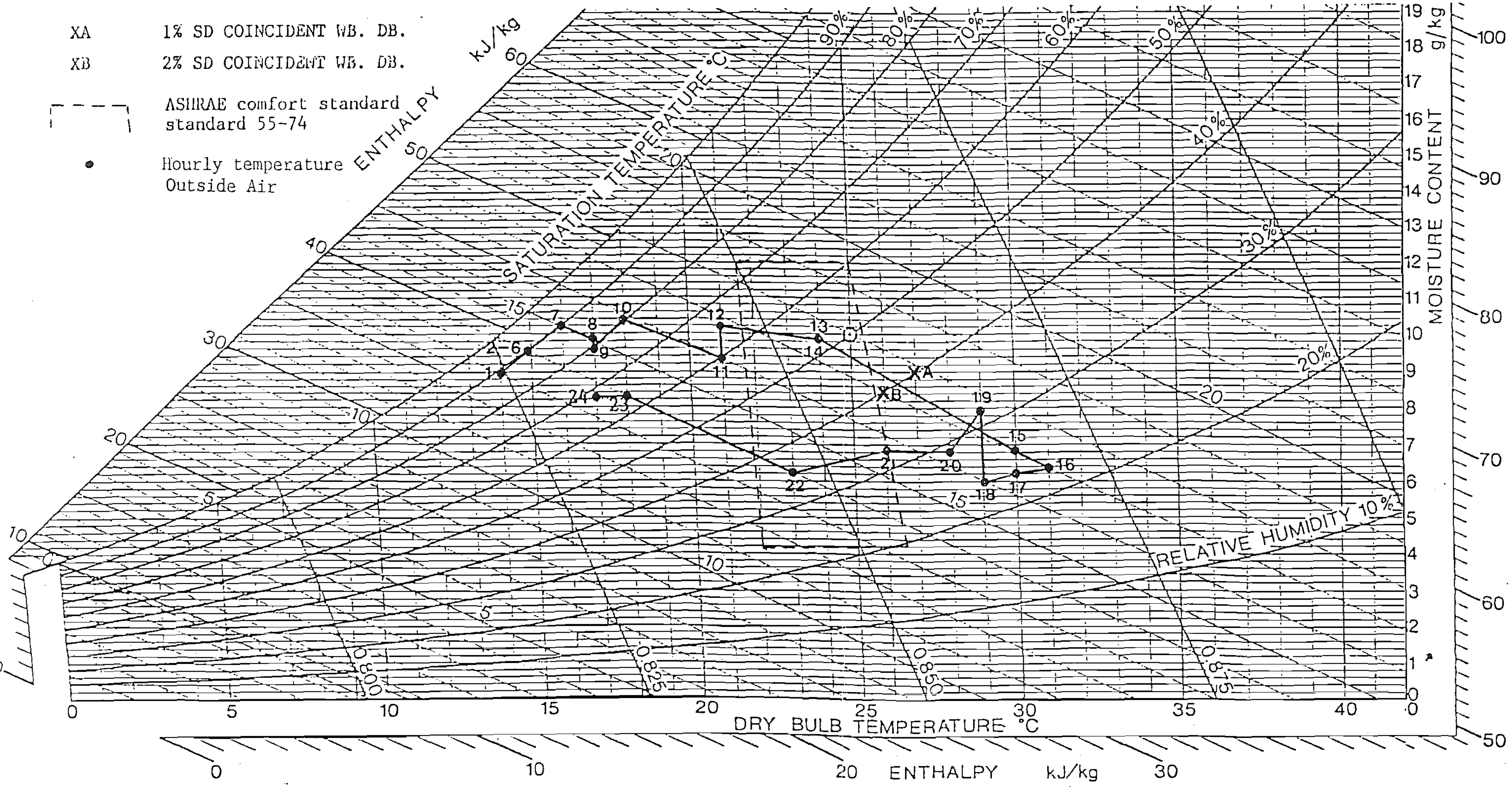


FIGURE 2.5 EVAPORATIVE COOLING TEMPERATURE CONSTRAINTS

> "Larger central cooling plants such as college campuses or other district cooling plants, represent another potential application. Occasionally central plants can take advantage of existing equipment in the process of retrofitting. A plant change from absorption chilling to compression chilling, for example, would make available a significantly oversized cooling tower that could work quite well as an evaporative chilling tower in conjunction with thermal energy storage capability." [Hatten M.J.1989]

The majority of air conditioning plant on campus is sized to operate on 12 deg.C artesian water, a temperature attainable by evaporative chilling for the majority of the year.

There will be periods during summer when the evaporative coolers will need to be supplemented by conventional chillers. With the use of thermal storage, night operation of the chillers, will minimise the costs of extra chilling capacity. As supplimentary cooling is required in summer only there should be no increase in electrical demand peak charges. Figure 2.6 shows a schematic diagram of an evaporative chiller with thermal storage system which allows topping up or backup with a conventional chiller.

There are several methods of achieving evaporative cooling of chilled water.

a)Cooling Tower. The open water cycle used here requires the use of heat exchanger between the cooling tower and the chilled water system.

b)Closed Circuit Evaporative Water Cooler. The closed water circuit used here removes the need for the external heat exchanger used for the cooling tower. Sludging, chemical dosing, and microbial problems are reduced significantly.

c) Strainer Cycle. This is a patented system allowing

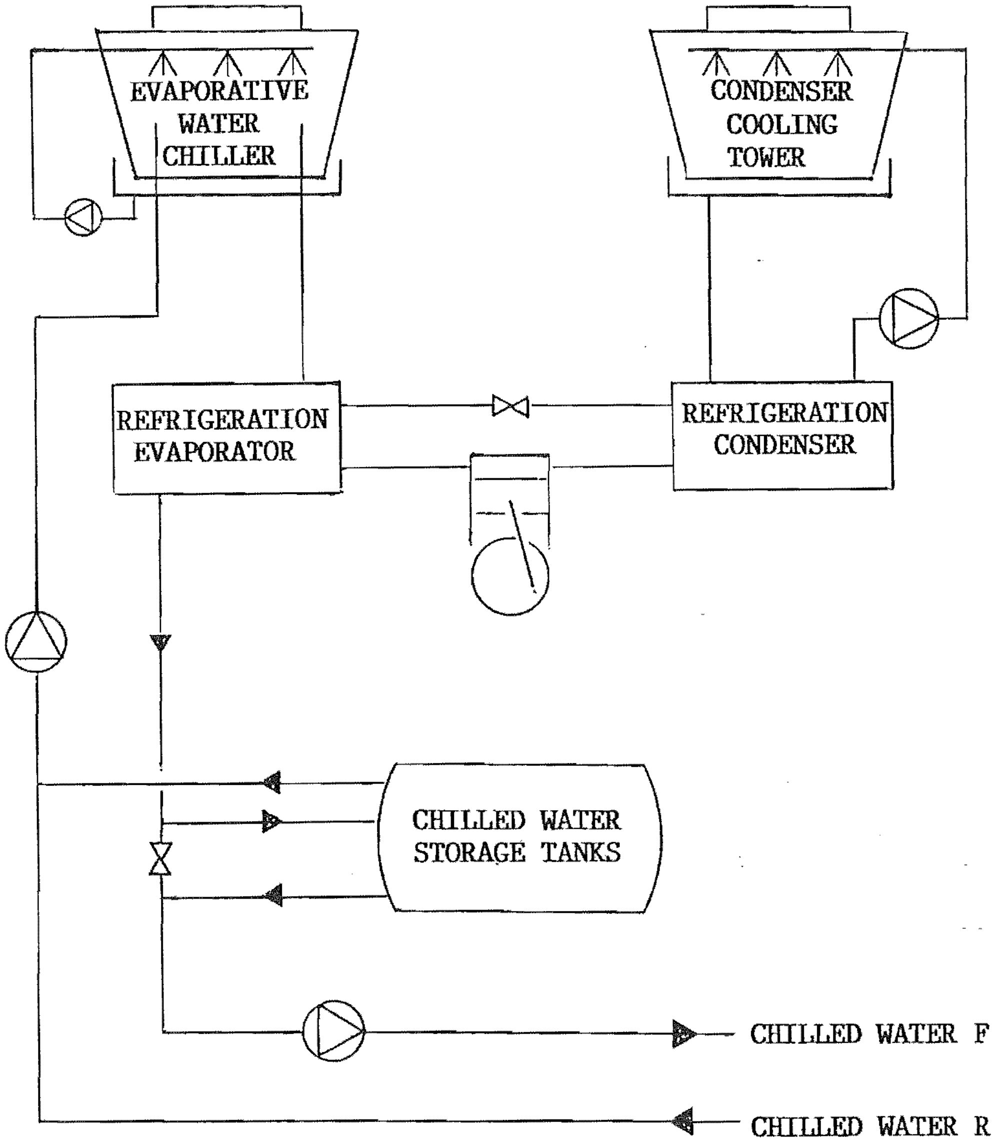


FIGURE 2.6

EVAPORATIVE CHILLING WITH THERMAL STORAGE direct use of a chillers cooling tower to provide direct evaporative cooling of chilled water. Capacity is however limited as the cooling tower will be sized for a greater water temperature difference due to the higher condensor temperature.

> "Chilled water storage and eutectic salt storage are applicable for direct evaporative chilling. In genera l, ice will not work as a thermal storage medium for e vaporative chilling because the working fluid temperat ures are lower than what is practical to achieve with a cooling tower."
>
> [Hatten M.J.& Johnston T.W.]

Because of the need for chemical dosing of cooling tower water to restrict microbial growth, indirect evaporative cooling using heat exchangers or operating the chiller as a heat exchanger is the preferred method of evaporative chilling.

Evaporative Cooling And Legionaires Disease.

Bacterial growth in open water systems in air conditioning systems is a significant problem, causing fouling, corrosion, and health hazards. It can be controlled by chemical dosing and careful maintenance, but these are understandably disliked. The extent of bacterial growth is determined by several factors. M. H. Lyons discusses the conditions required for bacterial growth;

> "The occurance of the main Legionaires Disease bacterium, legionela pneumophila, is common in nature but appears to be rare in properly treated cooling water systems. It has fairly specific requirements if we try to grow it in the laboratory and these are, a source of dissolved iron, a source of the aminoacid cystine, a pH between 6.6 and 7.2, and a temperature of between 20 and 50 deg. C."

As the water operating temperature in an evaporative cooler is not significantly above the wet bulb temperature of the air (which in Christchurch is at the lower limit of the bacterial growth temperature range) the chance of bacterial growth will be severly limited. Watt 1986 states that there are no reported incidents of Legionaires disease associated with evaporative coolers.

2.3.4 DISCUSSION - CAMPUS COOLING.

While operating cost savings are possible with centralised chiller plant, the installation costs are considerable, and without detailed data on the operating characteristics of air conditioning plant on campus a more accurate economic analysis is not possible.

The use of evaporative cooling on campus appears to be quite feasible. Combined with careful energy efficient design of new buildings it would be capable of providing cheap energy efficient cooling for buildings on the campus.

Similarly the use of evaporative chilling of cooling water in conjunction with a moderatly sized refrigeration cycle water chiller appears to be technically feasible and considerably cheaper than a conventional centralised or local water chiller installation.

In any case the operating costs will be higher than those with the current situation where the majority of cooling is derived by free cooling with artesian water.

2.4 THERMAL STORAGE SYSTEMS.

2.4.1 THERMAL STORAGE BENEFITS.

In order to maximise the use of cheaper off-peak electricity rates, thermal storage can provide the means of bridging the time lag between energy consumption when it is available cheaply, and the desired end use of that power, typically during daytime demand peaks.

The most obvious example of thermal storage in use is the common domestic hot water cylinder, which is invariably on a separately metered ripple controlled electricity supply, at a lower charge. Depending on the particular features of a thermal energy user, there are many thermal energy storage alternatives available.

2.4.2. Cooling Thermal Storage.

In light of the university's probable need for an energy efficient chilled water supply, analysis of the potential benefits of low temperature storage for the campus is important.

The main benefits of cool thermal storage are;

1. Thermal storage and load shifting can provide on peak cooling (or heating) at close to off peak costs. This advantage only applies to systems where the air conditioning load is coincident with the electrical demand peak charges. As electrical demand peak charges occur during winter in New Zealand, the argument for thermal cold storage in order to reduce peak electrical demand charges is contradicted. [STEWART L.J.]

However air conditioning loads with a low solar component may still have a significant duty during winter electrical demand peak periods, and thus benefit from cold thermal storage.

2. Reduced investment in refrigeration equipment as plant can be sized to average demand rather than peak demand.

3. An appreciable reduction in power cost can be achieved due to;

a. Increased operating efficiency of chilling plant as it runs at optimal design capacity continuously

b. Higher refrigeration COP of chillers run at night when ambient temperatures are lower.

c. The reduction in peak capacity, combined with the effect of shifting chilling load to off peak electricity periods, considerably reduces electrical demand charges, as well as the reduction in electrical energy consumption.

4. Because of the ability to shift the time of operation of the chilling plant the potential for operating the storage system as a heat pump providing useful heat for buildings exists.

The ability to use the reject heat off the chilling system is dependant on finding a suitable heat sink. With night rate electricity powered storage chillers the possibility of using the reject heat for pre-heating buildings for the following day is the most obvious solution.

Two main thermal storage systems exist, chilled water and ice storage.

2.4.3. Chilled Water Storage Systems.

Of the common materials available for thermal storage water has the highest specific heat capacity (4.18 kJ/kgK). As water is widely used as a heat transfer medium in building services it is a prime choice for use in thermal storage applications.

Chilled water for air conditioning normally has a flow temperature of about 6 deg C, therefore chilled water storage is required at or below this temperature. Return water from the air conditioner is at about 15 deg C. In order for the chilled water storage tank to be capable of storing any thermal capacity (or thermal energy) it must operate across a temperature differance equal to or greater than the differance between chilled water flow and return temperatures. Ideally two storage tanks are required, one to store chilled water and one to store the warmer return water. Chilled water is then stored by circulating warm returned water through a chiller to the cold tank, and the chilled water storage is used by circulating the chilled water from the cold tank through the air conditioning plant. Without separation of the cold and warm parts of the thermal store, blending of the cold and warm water reduces the temperature differance to zero, and with no temperature differential no energy storage is possible.

> "This leads us to the conclusion that the effectiveness of a chilled water storage system is directly dependant on the amount of temperature differential between the tanks. It would be best if we could devise a way to accomplish this with a single tank." [Trane Co 1976]

Several systems for maintaining the temperature differance in a water storage system are in use;

1. Temperature Stratification in a Single Tank.

Return water at 15 deg C can float above stored chilled water at 5 deg C due to its lower density. A thermocline, or zone with a steep temperature gradient, is present between the lighter warm water on top and the denser cold water below. The effectiveness of the stratification (and hence the resulting thermal storage effectiveness) is dependant on reducing turbulence within the tank, especially at the inlet and outlets, resulting in the use of large diffusers at these points. Also the tank shape is limited to narrow vertical tanks of constant cross-section. As the thermocline may increase to a depth of 2 meters this system is only really suitable for a tank of considerable vertical height.

2. Flexible Diaphragm in a single tank.

Here a flexible diaphragm takes the place of the thermocline. The only losses are leakage and the thermal conductivity through the diaphragm, as blending between the stratified regions is prevented by the diaphragm. In operation the diaphragms can split and wear, although small holes have a negligible effect on the system performance.

3. Multiple Tank Systems.

By storing the water in separate tanks a maximum temperature differential can be achieved. However this would double the total storage volume required. By compartmenting a large tank, or installing a large number of small tanks and circulating between individual tanks this can be avoided.

"If the number of compartments in a tank is assumed to be n, then the number of compartments filled with water is n-1: one compartment is empty. ...... The storage efficiency of such an arrangement depends on the number of compartments in each tank and is given by the formula;

$$(n-1)/n \times 100\% = E$$

Thus if the tank is divided into six compartments the efficiency is;

$$(6-1)/6 \times 100\% = 83.3\%$$ " [Linton KJ]

Figure 2.7 shows a typical multiple tank chilled water storage system

The value for efficiency given here describes the ratio of useful thermal storage capacity to the total tank volume.

Constructing a labyrinth of interconected compartments can achieve a reliable temperature differential between the inlet and outlet of the storage system with negligible mixing.

The main disadvantage of water based storage systems is the cost of the large storage tanks. If a suitable site can be found chilled water storage can be attractive especially as existing chiller plant can be directly connected to the storage system

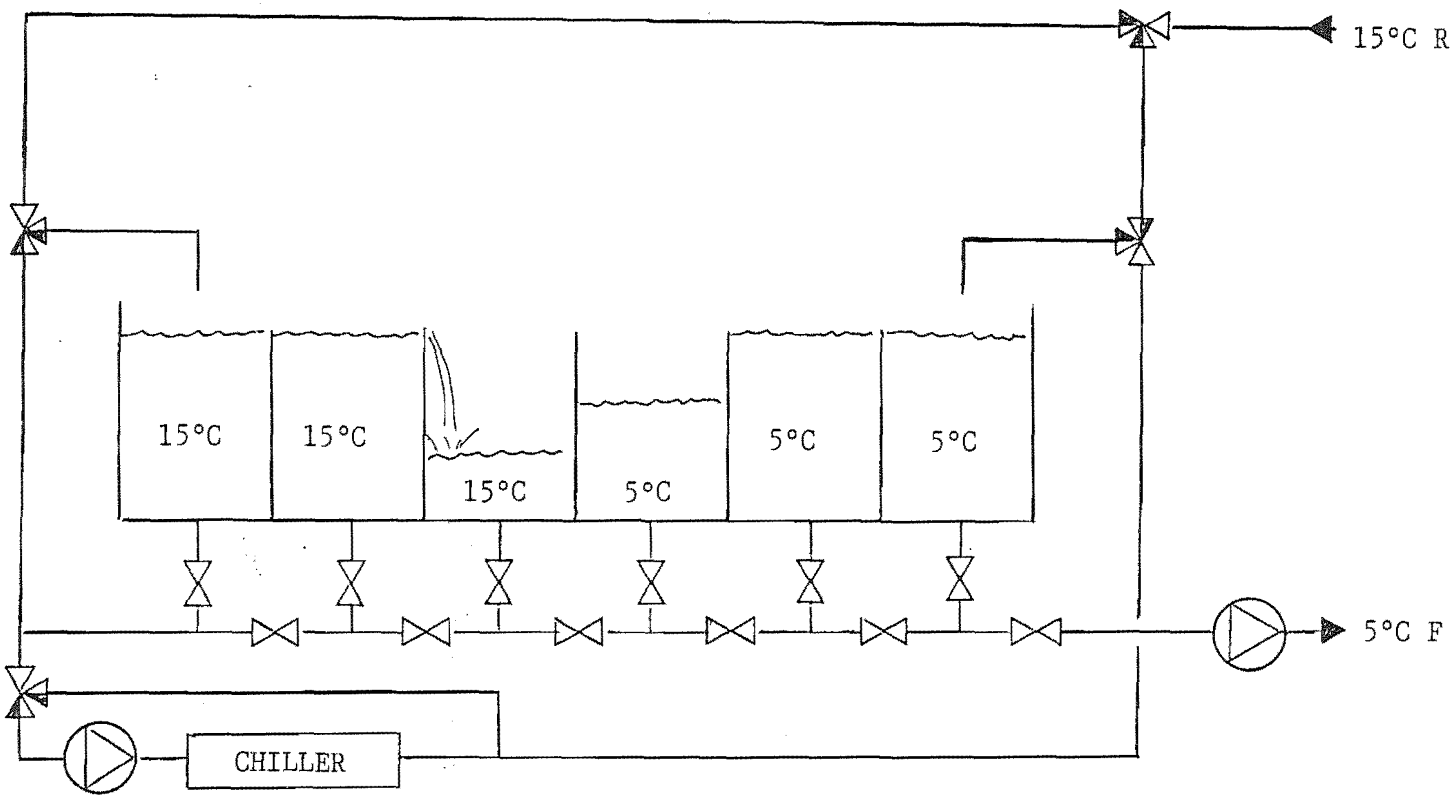


FIGURE 2.7 MULTIPLE TANK CHILLED WATER STORAGE SYSTEM

2.4.4. Ice Storage Systems.

In an ice storage system the ice is used for low temperature energy storage only, water chilled by the melting ice in the storage tank is used to transfer the stored cold energy to the air conditioning plant. By utilising the high latent heat of fusion (freezing) of water (335 kJ/kg) the volume required for storage is reduced considerably. For a given volume an ice store can store about ten times the energy of a chilled water store. As well as this the release of the thermal energy of the latent heat of melting ice occurs at the constant temperature of 0 deg C, allowing the problems of increasing chilled water flow temperature to be ignored.

In a new installation (where existing chillers are not available for connection to a chilled water storage system) ice storage is generally cheaper.

> "A lot of refrigeration can be purchased for the differance in storage tank costs. Ice storage tanks take up much less storage space volume - only about one tenth of the volume in this example. The storage system is less expensive and weighs a lot less. Essentially all cooling (energy) can be extracted from storage. With ice, problems of stratification, seperation, or sucking a tank dry etc are all avoided." [Stamm RH]

Ice storage system refrigeration plant suffers from a higher energy consumption for a given capacity when compared to chilled water. The increased energy consumption is due to the lower COP resulting from the lower evaporator temperature required to produce ice.

There are three main types of ice storage systems;

1. Ice on Coil - External Melt System.

Ice is built up on banks of serpentine coils submerged in an insulated tank of water. The refrigeration system builds up ice on the outside surface of the coils by circulating either, cold ethylene glycol brine through the coils, or by directly feeding low pressure refrigerant into the coil. When the ice has reached a designated thickness the refrigeration system is shut down. When chilled water is required the ice on the coils is melted by direct contact with the warm return water from the air conditioning plant. During this cycle the tank water is agitated by air bubbles from a low pressure distribution system beneath the coil in order to improve heat transfer. [Baltimore Air Coil]

Figure 2.8 shows a typical ice on coil system.

2. Ice on Coil - Internal Melt Systems.

This is similar to the ice on coil system described above. A glycol brine solution circulates throughout the entire cooling loop (from refrigeration chiller, through storage, to air conditioning plant load)and builds up, and melts the ice from the inside of the pipe on which it is built.

> "With this arrangement, the first ice to melt is that which is in immediate contact with the coil tubes. As the melt-out process continues, an annulus or ring of water develops between the tube and the ice, which continues to grow in thickness as the ice melts. The annulus provides an increasing resistance to the transfer of heat from the ethylene glycol to the ice, since the thermal conductivity of water is one - fourth that of ice. As a result of this design approach to internal melt, the ethylene glycol temperature rises with time.
>
> [Mc Cullough JM 1988]

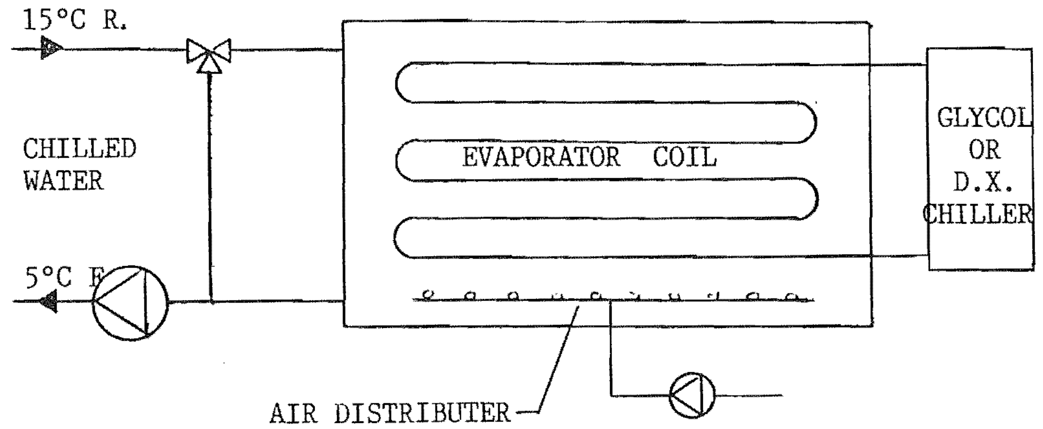
A variation of this system uses an "ice lens" system which attempts to overcome the heat transfer limitations by packing a large number of small water filled plastic containers into the storage tank.

3. Ice Harvester Systems.

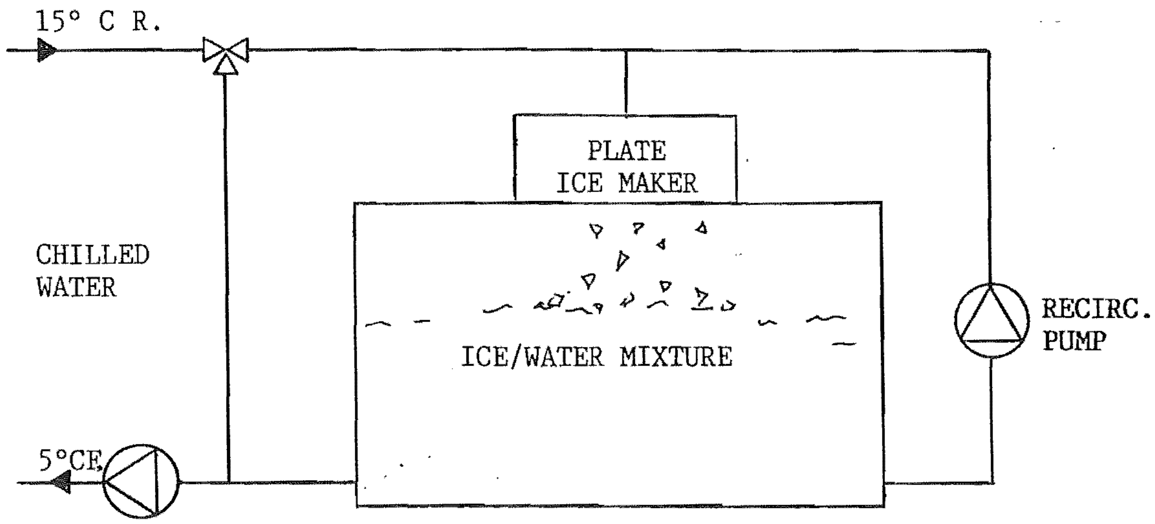
The ice making surface is typically suspended above the ice storage tank. The ice making surface consists of evaporator plates over which water flows. Ice forms on both sides of the plates in sheets about 6mm thick. At predetermined intervals hot refrigerant enters the plates causing the ice to break away and drop into the tank. The cycle is repeated as long as there is need for additional cooling reserves.

The advantages are that the ice harvester system gives a large heat transfer area to the circulating chilled water, allowing rapid melting with no risk of short circuiting of the return chilled water. This allows a continuous ice making capability which is impossible with systems where the coils are submerged in the ice storage tank, as the coil becomes encased in ice, insulating the heat transfer surface and reducing the efficiency of the system.

Figure 2.8 shows the operation of a typical ice harvester.



ICE ON COIL SYSTEM
(EXTERNAL MELT)



ICE HARVESTER SYSTEM

FIGURE 2.8 ICE STORAGE SYSTEMS

Ice Store System Configurations.

1.Full Storage.

The ice store is sized to provide the total daily cooling for the design cooling load. As the design load is based on the worst conditions, ie maximum building occupancy on the warmest expected day, the system will be severely under utilised for most of the year. Capital cost will be high, as in effect the system will be oversized by a factor of two, as the chiller or ice store alone will be capable of serving the design cooling load individually.

2.Partial Storage.

Both the chiller and store are run during the day to satisfy the cooling load, with the chiller charging the store at night. This results in an approximate reduction in chiller and ice store size of 50%.

Electrical energy consumption is reduced by a similar factor, and by operating the store only during demand peak and high energy cost periods, the chiller day operating costs will be held at a low level

This is a particularly suitable option for the university which faces its demand peak charges in the three winter months, when its cooling requirements are quite low.

Chiller/Store Arrangement.

With a full storage system the chilled water distribution system interfaces with the ice store only, however with a partial storage system both the chiller and ice store provide cooling, individually as well as simultaneously.

Three chiller/ice store arrangements exist;

1. Store Led Series Arrangement.

Refer to Figure 2.9a. Here the chiller has the highest possible suction temperature, and therefore the highest COP. Although the ice store gives off its cooling energy at constant temperature of 0 deg C, it faces a lower temperature restricting its melting capacity.

The store is charged by switching the three way valve to full bypass, (isolating the load from the store)

Full load operation has the three way valve on full flow with maximum chilled water temperature difference across the chiller and ice store. Part load operation has the valve modulating to provide a constant chilled water flow temperature. If constant flow temperature is not important eg if dehumidification is not neccessary, the control valve may be located in the chilled water return line maximising the return chilled water temperature to the chiller.

2.Chiller Led Series Arrangement.

Refer Figure 2.9b. With the chilled water already cooled by the ice store the chiller has a lower COP,than if in a store led configuration.

The ice store has improved cooling performance. Because of this, this system lends itself best to loads requiring a boost in cooling for only a short period. As the store is drawn down rapidly it may be quite small.

3.Parallel Arrangement.

Refer to Figure 2.9c. With the chiller and ice store in parrallel, output of each is controlled by individual valves.

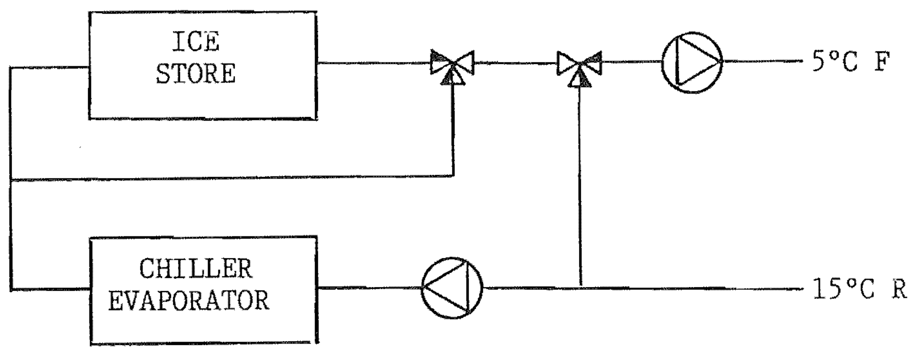


FIGURE 2.9a STORE LED SERIES ARRANGEMENT

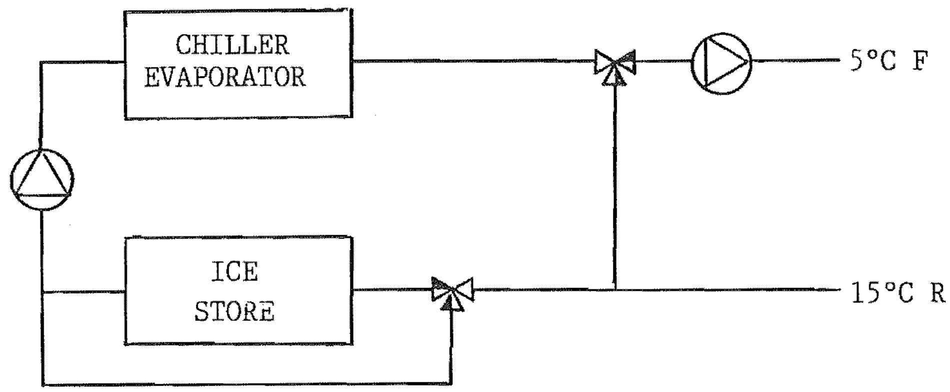


FIGURE 2.9b CHILLER LED SERIES ARRANGEMENT

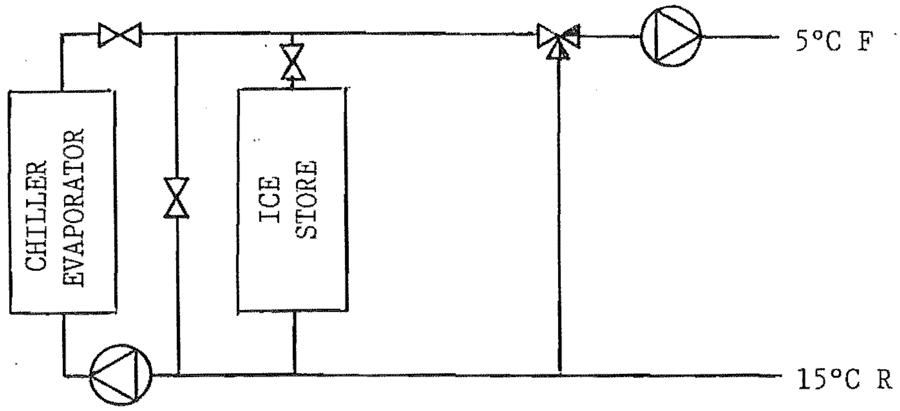


FIGURE 2.9c PARALLEL ARRANGEMENT

FIGURE 2.9 CHILLER - ICE STORE ARRANGEMENTS

Both the chiller and ice store face the same chilled water return temperatures maximising performance of each item. Under full load operation both the chiller and store provide maximum output. At part load the chiller operates at full output, with any excess of demand over the chiller output being passed to the store.

Ice Store Operation.

Refer to Figure 2.10.

1.Store Priority.

The ice store operates as the cooling base load while the chiller makes up any extra required capacity. The chiller operates at reduced capacity, resulting in a lower COP. The peak demand consumption of the chiller is indefinite but will be minimised. There is a tendency for both the chiller and ice store to be oversized with store priority, resulting in increased capital costs when compared with chiller priority. Store priority works best with a time of use electrical tariff as the chiller can be shut down during high electrical cost periods.

2.Chiller Priority.

Here preference is given to running the chiller at full load, which is sized to provide a daily base load as well as the night ice store charging duty. The ice stores output modulates to provide the peak cooling load.Ice store charging operation of the chiller is minimised, which is desirable as ice making incurs a loss of COP.

Where there is a peak demand function to the electrical tarrif this system is ideal as there little to be gained from

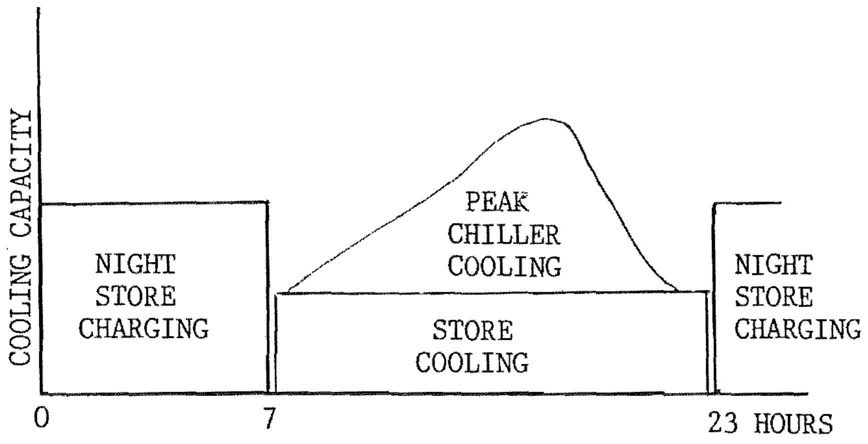


FIGURE 2.10 STORE PRIORITY OPERATION

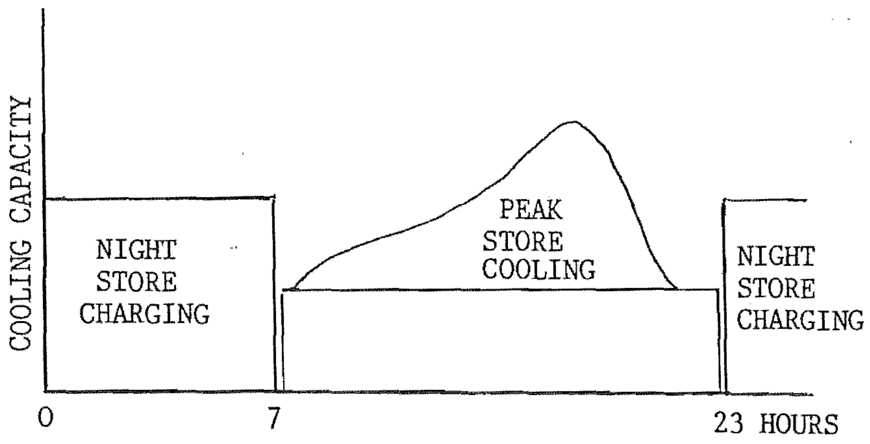


FIGURE 2.10b CHILLER PRIORITY OPERATION

FIGURE 2.10 PARTIAL ICE STORAGE OPERATION

using the chiller at less than full output at times when demand isn't at peak levels. This is especially applicable to the university's case where demand peaks are monitored for three months in winter, where providing the control system allowed the ice store to meet the total cooling load during winter demand peaks, and shut off the chiller in winter, the chiller priority system would be most applicable for operation for the rest of the year.

Ice Storage System Sizing And Selection.

In order to achieve any of the benefits of an ice storage system both chiller and ice store size must be optimised for the cooling loads and frequency of those loads,the type of store and chiller configuration, the electrical tariff structure, and the control strategy.

2.4.5. Ice Store Operating Strategy For The Campus

Sizing and Selection.

1. Size the chiller and ice store for chiller priority operation based on air conditioning cooling load design conditions in March. This will achieve;

a) Peak cooling loads will be satisfied by the chiller and ice store operating together in chiller priority, when normal term occupancy is coincident with the hotest weather conditions.

b) During periods when the cooling load is reduced eg. spring/autumn when a significant part of the cooling load is achieved by free cooling, operation on either store or chiller alone to make up any extra load depending on energy costs at that time.

c) During winter energy demand charge periods the ice store alone should cope with any balance of cooling loads over that satisfied by free cooling.

The chiller priority sizing will ensure minimum capital cost of chiller and ice store.

The above design assumes that maximum benefit of free cooling is achieved by fresh air / recirculated air control by both temperature and enthalpy control. All spaces should be pre cooled before occupancy.

When simple control is used on the storage system the system can only be optimised for one load pattern/energy tarriff structure. However the campus has an EMCS capable of making dynamic plant operating decisions based on immediate cooling load and energy tarriff conditions. For this reason parrallel conection of the chiller and store, will best allow optimum chiller COP and ice store meltout to be attained regardless of the EMCS selected method of operation.

Operation.

a) Summer.

During summer vacations any lecture theatres in use will be booked on the EMCS so the EMCS can determine the required level of ice to be built up enabling store only operation without excessive ice building,or day chiller operation. The decision on whether to make ice and use the store on the next day or run the chiller during the day depends on the ratio of chiller day COP to night COP and energy costs.

ie if COP night ice making < ($/kWh d / $/kWh n)*COP day then it will be cheaper to run the chiller on chiller priority during the day

Currently the summer night/day energy cost is 3.38/4.61 or 0.73. So if the chiller COP at night is lower than 0.73 of the chiller day COP then the system should be run on the chiller during the day.

During summer term time eg March, the system will run on chiller priority, but may run on store depending on the pricing differential between summer day and night rate energy costs.

b) Spring/Autumn.

Where possible cooling will be achieved by free cooling using incoming fresh air. Any balance required can be made up by the storage system running on chiller priority or store priority depending on thenight/day energy pricing differential.

c) Winter.

As for Spring Autumn operation free cooling should make up most of the cooling load.Any balance will be made up by the storage system running on chiller or store depending on the night/day energy pricing differential.
However during the months of June to August the system will operate on store only during periods when the EMCS determines demand peak charges are likely to be incurred. The winter pricing differential may be such that the store only will be run during the day. It is likely that the store alone will satisfy winter cooling loads.

2.4.6. Alternative Systems.

Binary Fluids.

A binary fluid is a mixture of fresh water and another fluid, typically ethylene glycol. When the solution is chilled the fresh water is frozen out of solution, forming discrete ice crystals, resulting in a binary fluid slurry which is pumped to storage, where a porous ice pack forms. Because of the high surface area of the ice crystal pack the binary fluid storage system can discharge its stored energy rapidly similarly to the plate ice harvester.

2.4.7. HEATING THERMAL STORAGE.

As virtually all of the energy used for heating on campus (both space heating and domestic hot water supply) is derived from the steam district heating system the scope for improvements by using heating thermal storage is limited. Because the steam is produced from coal fired boilers which are in operation all year round there is little need for storage, as the coal can be burnt at any time with no loss of thermal efficiency or cost penalty.

However this situation would be quite differant if an alternate strategy for boiler operation was introduced. If the boilers and steam district heating system were in operation only during the winter heating season, the requirement for domestic hot water would be best satisfied by night rate electric water heating storage tanks. There is also some question as to the ability of the existing boiler plant to meet expected growth in steam demand with expansion of the campus buildings. Thermal storage can play a role here in offsetting peak building steam demands, and prevent the need for increased boiler capacity.

Phase change materials (PCMs) and Eutectics use the latent heat of phase change of various materials for thermal storage at a wide range of temperatures. However these are used only in a limited number of systems overseas and unlikely to be commercially acceptable in New Zealand.

2.4.8. DISCUSSION - THERMAL STORAGE SYSTEMS.

With a cost of over $500,000 for an ice storage system capable of supplying cooling for about half the campus, and a similar cost for chilled water storage, assuming sufficient space can be found, either of the cooling storage systems are likely to be economically untenable. As with the district cooling schemes the high capital cost is unlikely to be recovered in savings. Conventional chillers would have lower capital costs and while still using energy at peak day time periods, may still be cheaper to operate in the long term if maintenance for the storage systems proves expensive.

The ability of evaporative cooling to provide inexpensive energy efficient air conditioning for the campus deserves further consideration as it avoids the high capital costs of the alternative schemes, and has low operating costs. In some cases evaporative coolers may be easily installed in existing air conditioning and ventilation plants.

2.5 CONVERSION OF STEAM TO HOT WATER HEATING.

2.5.1 STEAM TO HOT WATER CONVERSIONS.

Operating cost reductions of the order of an estimated 20% (actual saving 40%) are reported by Yetman, for the conversion from steam to hot water heating. Similarly Cole 1989 describes hot water conversion of three west coast hospitals where energy usage reductions ranging from 27% to 45% were achieved. The motives for conversion were;

1. Boiler age and condition
2. Difficulty in finding certified boiler attendants
3. Energy cost savings
4. Salary reductions through elimination of the requirement for full time boiler attendants

As well Yetman claims reductions of 40 and 50% in capital cost for hot water boilers compared to steam boilers. For the university to replace its existing steam boilers it would face a cost of $M2.6 for steam boilers, (source Fig 5.1 The Potential For Cogeneration Of Heat and Electricity) or about $M1.3 for hot water boilers, using Yetman's claim of 50% cost of steam boilers for hot water boilers.

Improvements in thermal efficiency of boiler plant are due to many factors, and while the physical design of shell boilers (whether steam or hot water) has not altered significantly, modern boiler control systems and operational methods have improved boiler efficiencies.

The main advantages of hot water over steam heating are:

1) Thermal Inertia. A large volume of water is constantly circulated around the heating system mains, and

this acts as a thermal flywheel damping out fluctuations in demand. At present the campus steam boilers have problems following the demand for steam as it varies throughout the day. The existing shell boilers contain a large volume of water and have chain grate stokers, resulting in a high thermal inertia boiler system. Altering boiler output significantly takes about 30 minutes especially if there is a deep bed of coal burning on the stokers. As steam loads are shut off or started up on campus the steam pressure fluctuates faster than the boilers can keep up with, occasionally blowing safety valves.

2) Lower System Heat Loss. The closed hot water system eliminates the losses associated with leaking steam traps, valve and pump glands, boiler blow down, and open condensate receiver flash loss and hot well losses.

3) Reduced Maintenance. Elimination of steam traps, condensate return systems, pressure control valves, safety valves, etc significantly reduces heating system maintenance requirements.

4) Improved Control. Steam system control is complicated by the fluctuation in steam pressure as well as flow rate as loads vary. For hot water systems the return temperature only varies, as the hot water system operates with constant flow rate and output temperature.

5) Economics. Higher system efficiencies are claimed for hot water systems, as well as reduced operating costs, maintenance and downtime.

Pressure and Temperature Constraints

In order to achieve flow temperatures above say 85 degC without boiling the hot water system must be pressurised. The flow temperature limit is determined by the saturation pressure of the system with a flash margin added as a factor of safety. For the campus system pressurisation could be achieved by either a static head due to a system feed tank located on the roof top of say the Physics building, or by using a gas pressurised expansion vessel. The system feed tank is simplest and would give a system upper temperature of about 120 degC. This is based on a building height of 30m giving a static head of 3 Bar with corresponding saturation temperature of 133degC, and subtracting a 15deg flash margin, to give a system upper limit of 118degC. As the amount of energy transmitted is a function of the flow and return temperature difference, a return temperature at least 20 deg below the flow temperature is desirable.

Another approach is to match the hot water flow temperature to the saturation temperature of the existing steam system ie 165degC (from sat.steam @ 7 bar). Existing heat exchangers and calorifiers would not be derated significantly if this was done. The system would have to be pressurised by a pressurised expansion vessel.

Steam / Hot Water Operating differences

A simple analysis of both the existing steam heating and hot water systems gives the following results:

1) System Efficiency. Currently the steam heating system is operating at an estimated efficiency of 65% [source, main-

tenance department staff] during the winter heating season. At other times of the year as the heating load reduces, while circulation and operational losses remain constant, the efficiency drops even further.

2) System Leakage. 95% of condensate (at 75degC) is returned to the boiler hot well with the existing steam system. Also [Beal 1987] gives a steam system leakage rate of 5.0% and hot water system leakage rate of 0.5%. The hot water system circulates approximately 10 times the mass of water of the steam system so the volume lost is about the same, however the steam enthalpy is greater than the hot water enthalpy.

3) Boiler Blowdown. The quantity of boiler water lost during blowdown is difficult to determine as blowdown is intermittent, with no heat or flash recovery used. Blowdown rate, whether continous or intermittent, is determined by the percentage of dissolved solids in the boiler, so for a given boiler installation the blowdown rate should theoretically be the same whether continous or intermittent blowdown is used. Beal gives a continuous blow down rate of 2% of steam flow for a boiler installation with 95% condensate return.

4) Feed and Circulation Pumps. As the campus steam system uses Weir steam driven boiler feed pumps, and there are no extra pumping costs associated with the steam system, the pumping energy requirements are part of the boilers net energy consumption. The hot water system will require electric boiler feed pumps and system circulating pumps for each heating main. Main circulating pumps are estimated to have

the following duties;

HEATING MAINS	FLOW RATE	HEAD LOSS	PUMP MOTOR
Engineering	11kg/s	95kPa	2.5kW
Science	23kg/s	93kPa	4 kW
Arts	38kg/s	310kPa	18 kW

Appendix E contains analysis of the operation of both steam and hot water heating based on a design heating load derived from the present campus annual heating energy consumption. The results given below are largely comparative as the annual mean design load doesn't take into account the effect of varying heat loads and plant efficiencies during the year on the two systems.

2.5.2 STEAM/HOT WATER HEATING ANALYSIS RESULTS.

Annual Operating Costs for Steam System.

Fuel 4390 tonnes @ $88/tonne.	$386,320
Water treatment $5000/ann.	$5,000
Boiler operators 5 * $80,000	$400,000
Steam system maintenance assume 2.5 Fitters @ $30,000 * 2	$150,000
	$941,320

Note.

As exhaust from the Weir boiler feed pumps is used for heating the hot well, and the Weir pump exhaust losses are felt to be recovered by the improved boiler feed water condition they are neglected.

Boiler operator salaries are taken at the true cost to the university, that is twice the gross salary paid.

Annual Operating Costs for Hot Water System.

Fuel 3862 tonnes @ $88/tonne.	$339,912
Water treatment - negligible	
Boiler operators 1 * $80,000	$80,000
Pumping operating costs	$4220
Steam system maintenance assume 0.5 Fitters @ $30,000 * 2	$30,000
	$454,132

With the hot water system, coal consumption is reduced to 87% of that required by the present steam system. This is a conservative figure which doesn't take into account;

1. Reductions in pipe heat loss. The hot water system

will only have about 80% of the heat loss of the steam system.

2. The improvement in load control and reduced losses at the loads will reduce the design load.

Experience with conversions (see Yetman) shows the actual savings to be higher than the estimated savings.

The significant reduction of labour costs in the operating cost of the steam heating system is the most important saving in conversion to hot water heating. As the hot water plant is capable of being operated without any boiler attendants, the overall operational cost of the hot water system may be below half that of the steam system. However one boiler operator has been included in this analysis as operation of ancillary boilerhouse plant such as coal and ash handling equipment will still be required. The hot water system operating costs are about 59% of those of the steam system.

The costs of conversion to hot water operation can be estimated as follows:

New hot water shell boilers:			$M1.3
Alternately, use existing boilers for hot water, accepting some derating.			$100,000
New Heating mains:			
Engineering.	560m		$56,000
Science.	910m		$110,000
Arts.	1220m		$145,000
Additional Items:			
Pumps,etc		$50,000	
Alterations to system:			$300,000
Total, new boilers,			$M1.9
Total, existing boilers,			$660,000

Notes:

1. Any derating of the existing boilers (which is unlikely) may be compensated by the extra thermal storage created by the mass of water in the heating mains, (approx 45 tonne in the mains alone).

2. Heating main costs from Rawlinsons New Zealand

Construction Handbook.

3. The existing steam flow mains are adequately sized to carry the hot water flow or return. If this pipe work was reused the heating main alteration cost would be halved.

By using the existing boilers as hot water boilers, (there should be no reason why this cannot be done, although some internal modifications to ensure good circulation may be required) the conversion payback period may be of the order of 1.5 years. Having to replace the boilers will produce a payback period of about 4.5 years. These figures are rough orders of cost only, but without a detailed investigation into the conversion costs no better estimate is possible. However as an indication of the economic viability of this type of conversion the savings could produce an attractive return.

2.6.CONCLUSIONS.

Of the energy systems discussed in this paper only two would appear to provide a net benefit for the university. The use of evaporative cooling for air conditioning and conversion from steam to hot water heating are both capable of giving the university an attractive net return on the capital investment required to install the plant. This is as well as providing services to the university at lower cost and lower energy consumption than the existing systems. The investigations carried out in this thesis have been of a general nature giving an overview of the performance and costs of various energy technologies. In all cases the performance and costs of specific models and plant applications will vary from the results determined, and further engineering and economic analysis will be required.

The high capital cost of CHP plant in relation to the relativly low current energy costs, ruins the economic feasibility of a conversion to CHP production for the campus.

However the analysis does demonstrate the thermodynamic and economic effectiveness of gas turbine CHP plant. In a situation where a new complex, such as the campus was being set up with a suitable piped gas supply, it is likely that detailed analysis would show the gas turbine CHP plant may (in the long term) be a more competitive option than the conventional separate electrical and heat supply. (The analysis used in this study didn't take into account the installation costs of separate heat and power supplies). At the time of completion of this thesis, General Electric were offering a commercially ready steam injected gas turbine, with at least 40% operating thermal efficiency, which would provide the benefits discussed in the section on the Cheng cycle gas turbine, and may be more cost effective than the installation of seperate heat and power supply.

The steam based CHP schemes are economically less viable, however in the South Island of New Zealand they may be acceptable providing the owner was prepared to accept a longer pay back period required to recover the high capital cost. As with the gas turbine the case of the CHP plant following the heat load provided better payback and higher operating efficiency.

The use of centralised chilling plant and thermal storage is uneconomic as the capital costs are high, and the savings low due to the relatively benign temperate climate in which the university is located. Because of this, evaporative cooling shows good potential for satisfying cooling loads on the campus at very low capital and operating costs.

A trend seen in all the cases was that high capital cost is difficult to recover and is one of the main threats to the viability of new energy efficient systems projects.

APPENDIX. A.

Back Pressure Turbine Efficiency Calculation

Turbine Entry Condition;
40bar 400deg Steam,hg = 3215 kJ/kg
Turbine Outlet Condition; 7bar, hout(s) = hg = 2790 kJ/kg
Turbine isentropic Efficiency; Eisen.= 0.9
Turbine Exhaust enthalpy;
hout = hin - Eisen * (hin - hout(s))
= 3215 - 0.9 (3215 - 2790) = 2832 kJ/kg.

For 3MW output, steam flow given by;

$$\frac{3MW * 1000}{(3215 - 2832)} = 7.833 \text{ kg/s}$$

Heat rejected; (Boiler Feed @100deg C, hf = 419 kJ/kg)
7.833 kg/s * (2832 - 419) = 19 MW

Heat Input;
(3215 - 419) * 7.833 kg/s = 21.9 MW

Design Peak Output Thermal Efficiency;

$$\frac{3 \text{ MW}}{21.9 \text{ MW}} = 0.137 = 13.7\%$$

Design Work / Heat ratio;

$$\frac{3 \text{ MW}}{19 \text{ MW}} = 0.158$$

KEARTON in Steam Turbine Theory and Practice, pg 398, gives 0.14 as the ratio of no load steam consumption, to full load steam consumption. This gives a no load steam consumption of;

7.833 kg/s * 0.14 = 1.09 kg/s = 3.06 MW

The efficiency function used in the CHP analysis spreadsheet is produced by linear regression of the performance data resulting from this calculation. The no load steam consumption value is the y axis intercept value and the slope is determined from the relationship of steam energy input to, and power output (for the case when the CHP plant follows the electrical load), or heat output (for the case when the CHP plant follows the heat load) for the CHP plant.

For the CHP plant following the electrical load:
No load steam input = 3.06 MW
Steam input / electrical load = 6.28 MW/MW

This gives the efficiency function for the back pressure turbine following the electrical load as:
Qin = 3.06 + 6.28 * Wout MW

As the performance of the steam turbine following the heat load is required, and the heat load is independant of the electrical load, a seperate thermal efficiency function must be developed for the CHP plant following the heat load.

For the CHP plant following the heat load:

No load steam input	=	-0.57 MW
Steam input / heat load	=	1.189 MW/MW

This gives the efficiency function for the back pressure turbine following the heat load as:

Qin = 1.189*Qout - 0.57MW MW

APPENDIX B
PASS OUT TURBINE EFFICIENCY CALCULATION

Turbine Entry Condition;
40bar 400deg, hin = hg = 3215 kJ/kg
Turbine Passout Condition; 7bar,
hpo(s) = hg = 2764 kJ/kg
Turbine Exhaust Condition;
0.1bar hex(s) = hg = 2145 kJ/kg
Turbine isentropic Efficiency; 0.9

Pass Out Steam enthalpy (hpo);
hpo = hin - Eisen. * (hin - hpo(s))
= 3215 - 0.9 (3215 - 2764) = 2809 kJ/kg.

Exhaust Steam enthalpy (hex);
hex = hpo - Eisen. * (hpo - hex(s))
= 2809 - 0.9 (2809 - 2145) = 2211 kJ/kg.

Inlet Stage enthalpy change;
Delta hin = hin - hpo
= 3215 - 2809 = 406 kJ/kg
Exhaust Stage enthalpy change;
Delta hex = hpo - hex
= 2809 - 2211 = 597 kj/kg

<u>Operating Condition - No Extraction.</u>

For 3MW output, steam consumption given by;

$$\frac{3MW * 1000}{(3215 - 2211)} = 2.98 \text{ kg/s}$$

Heat rejected; (Boiler Feed @100deg C, hf = 419 kJ/kg)
2.988 kg/s * (2211 - 419) = 5.35 MW

Heat Input;
(3215 - 419) * 2.988 kg/s = 8.35 MW

Design Peak Output Thermal Efficiency;

$$\frac{3 \text{ MW}}{8.35 \text{ MW}} = 0.359 = 35.9\%$$

KEARTON in Steam Turbine Theory and Practice, pg 398, gives 0.12 as the ratio of, no load steam consumption to full load steam consumption for a pass out turbine with no extraction. This gives a no load steam consumption of;
2.988 kg/s * 0.12 = 0.359 kg/s
0.359 kg/s * (3215 - 419) = 1.00 MW

From the above calculated values the "no extraction" steam consumption line can be produced on figure 2.3

<u>Operating Condition - Full Extraction</u>

For 3MW output, steam consumption given by;

$$\frac{3MW * 1000}{(3215 - 2809)} = 7.389 \text{ kg/s}$$

Heat input;

$$7.389 \text{ kg/s} * (3215 - 419) = 20.66 \text{ MW}$$

Heat rejected at pass out pressure;

$$7.389 \text{ kg/s} * (2809 - 419) = 17.66 \text{ MW}$$

Design peak output Thermal efficiency;

$$\frac{3 \text{ MW}}{20.66} = 0.145 = 14.5\%$$

Design peak work / heat ratio;

$$\frac{3 \text{ MW}}{17.66} = 0.17$$

KEARTON in Steam Turbine Theory and Practice, pg 398, gives 0.14 as the ratio of, no load steam consumption to full load steam consumption for a pass out turbine with full extraction. This gives a no load steam consumption of;

$$7.389 \text{ kg/s} * 0.14 = 1.034 \text{ kg/s}$$

$$1.034 \text{ kg/s} * (3215 - 419) = 2.89 \text{ MW}$$

From the above calculated values the "full extraction" steam consumption line can be produced on figure 2.3.

Intermediate condition - partial extraction

Here the power developed in the low pressure turbine stage is the same as the no extraction case for a given power output. The passout steam flow will generate additional power as it proceeds through the high pressure stage, depending on the amount of pass out bleed steam determined by the heating load.

Lines of constant pass out heat output can be interpolated between the no extraction and full extraction lines on the steam consumption diagram.

The efficiency function used in the CHP analysis spreadsheet is made up from two parts produced by linear regression of the heat and electrical loads, ie

Qin = No Load steam + Pass out steam + Elec. load steam

The no load steam consumption value is the y axis intercept value of the no extraction line. ie here no pass out heat is released from the turbine intermediate stage and the turbine acts as condensing turbine.

The slope of the pass out steam consumption component is determined from the relationship of steam energy input to the pass out heat output (see "Pass Out Heat" lines on Figure 2.3). The amount of pass out heat effectively raises the no load steam consumption value up the steam consumption axis to the y axis intercept value for the relevant passout heat load.

The effect of the electrical load on the CHP plant is given by the relationship of steam consumption to electrical load for the simultaneous pass out heat load.

No load steam consumption at No extraction:
No Load steam input = 1.12 MW

Pass out heat component:
Steam input / pass out heat = (Qout * 0.684) MW/MW

Electrical load component:
Steam input / electrical load = (Wout * 2.433) MW/MW

These give the steam input for a pass out steam turbine CHP plant, modulating its steam input and pass out steam to simultaneouisly match both heat and electrical loads, as the following equation;

$$Qin = 1.12 + (Qout * 0.684) + (Wout * 2.433)$$

This is used to determine the steam energy requirements for the pass out turbine in the spreadsheet.

APPENDIX C
GAS TURBINE EFFICIENCY CALCULATION.

Performance data for the Allison 501k gas turbine driven generator set from;
General Specification Gas Turbine Generator Set,
General Motors Model GM 501KB.

For continuous rating 3125kW electric output;

Specific fuel consumption	100% output	= 313.3gr/hr
	50% output	= 411.0gr/hr

At rated continous output, Thermal Efficiency = 28.4%
At 50% continuous output,

$$\text{Thermal Efficiency} = \frac{313.3}{411.0} * 28.4\% = 21\%$$

Exhaust temperature 514 deg C

For ease of use in the spreadsheet it is assumed that the gas turbine's thermal efficiency varies linearly with output.

Linear Regression on the above power output and thermal efficiency values give the following;

No load thermal efficiency	14%
Specific change in efficiency	4.67% / MWe

The Thermal Efficiency function for the gas turbine based CHP plant following electrical load is then;

$$\text{Effy th} = 14\% + 4.67\% * (\text{Wout MW})$$

As the performance of the gas turbine following the heat load is required, and the heat load is independant of the electrical load, a seperate thermal efficiency function must be developed for the CHP plant following the heat load.

At rated output $\quad Q_{in} = \dfrac{3.125\text{MWe}}{28.4\%\ \text{effy}} = 11.0\text{MW}$

at $Q_{out} = 11.0 - 3.125 = 7.875\text{MW}$, Effy = 28.4%

At 50% rated output $\quad Q_{in} = \dfrac{1.50\text{MWe}}{21.0\%\ \text{effy}} = 7.14\text{MW}$

at $Q_{out} = 7.14 - 1.50 = 5.64\text{MW}$, Effy = 21%

Linear regression on the above values of heat output and thermal efficiency give;

Thermal efficiency at no heat load	= 7.32%
Specific change in efficiency	= 1.91%

The Thermal Efficiency function for the gas turbine based CHP plant following heat load is;

$$\text{Effy th} = 7.32\% + 1.91\% * (\text{Qout MW})$$

APPENDIX D. CHP PERFORMANCE ANALYSIS SPREADSHEET.

THERMAL LOAD FILE

TIME hrs	Winter Peak Steam Load MW	Winter weekday Steam Load MW	Winter weekend Steam Load MW	Summer weekday Steam load MW	Summer weekend Steam Load MW
1.00	5.80	5.80	3.60	2.60	2.60
2.00	5.80	5.80	3.60	2.60	2.60
3.00	6.90	6.90	3.60	2.60	2.60
4.00	7.20	7.20	3.90	2.60	2.60
5.00	11.50	11.50	4.70	3.20	2.60
6.00	14.00	14.00	4.70	3.60	2.60
7.00	14.30	14.30	4.70	4.20	2.60
8.00	12.70	12.70	4.40	3.90	2.60
9.00	11.10	11.10	4.20	4.20	2.60
10.00	9.60	9.60	4.40	3.90	2.60
11.00	8.20	8.20	4.20	3.40	2.60
12.00	8.20	8.20	4.20	2.90	2.60
13.00	8.20	8.20	4.20	2.90	2.60
14.00	8.20	8.20	4.20	2.90	2.60
15.00	8.20	8.20	4.20	2.90	2.60
16.00	8.20	8.20	4.20	2.60	2.60
17.00	6.80	6.80	4.20	2.60	2.60
18.00	6.60	6.60	4.20	2.60	2.60
19.00	6.60	6.60	4.20	2.60	2.60
20.00	6.60	6.60	4.20	2.60	2.60
21.00	6.60	6.60	3.90	2.60	2.60
22.00	6.30	6.30	3.90	2.60	2.60
23.00	5.80	5.80	3.60	2.60	2.60
24.00	5.80	5.80	3.60	2.60	2.60

ELECT'L LOAD FILE

TIME hrs	Winter weekday Elect'l Load MW	Winter weekend Elect'l Load MW	Summer weekday Elect'l Load MW	Summer weekend Elect'l Load MW
1.00	0.80	0.60	0.50	0.50
2.00	0.80	0.60	0.50	0.50
3.00	0.80	0.60	0.56	0.50
4.00	0.90	0.60	0.70	0.50
5.00	1.00	0.60	0.70	0.50
6.00	1.10	0.60	0.70	0.50
7.00	1.20	0.65	0.70	0.50
8.00	1.50	0.70	1.20	0.60
9.00	2.30	0.75	1.50	0.76
10.00	2.70	0.90	1.60	0.80
11.00	2.90	0.98	1.60	0.80
12.00	3.00	0.95	1.55	0.60
13.00	2.70	0.90	1.60	0.60
14.00	2.70	0.90	1.60	0.65
15.00	2.90	0.94	1.60	0.65
16.00	2.80	0.97	1.40	0.65
17.00	2.40	0.93	0.90	0.68
18.00	2.00	0.95	0.75	0.66
19.00	2.00	0.80	0.70	0.70
20.00	1.80	0.84	0.60	0.66
21.00	1.50	0.80	0.60	0.70
22.00	1.30	0.80	0.50	0.70
23.00	1.10	0.76	0.50	0.60
24.00	0.80	0.65	0.50	0.56

N.B. The following performance analysis summary uses these abbreviations

WWD.S = Winter weekday daily steam load
WWE.S = Winter weekend daily steam load
SWD.S = Summer weekday daily steam load
SWE.S = Summer weekend daily steam load
WWD.E = Winter weekday daily electrical load
WWE.E = Winter weekend daily electrical load
SWD.E = Summer weekday daily electrical load
SWE.E = Summer weekend daily electrical load

Qin = Heat input to turbine
Qout = Heat output from turbine
Wout = Work (electricity) output of turbine
Wbal = Work (electricity) balance
Win = Work (electricity) imported

CHP PERFORMANCE ANALYSIS.
GAS TURBINE FOLLOWS HEAT LOAD.

TIME hrs	Winter weekday Steam Load MW	Thermal Eff'y of CHP plant	Qin MW	Wout MW	Winter weekend Elect'l Load MW	Wbal MW
1.00	5.80	WWD.S*1.91+7.32	WWD.S/(1-EFF/100)	Qin - WWD.S	0.80	Wout - WWD.E
2.00	5.80	WWD.S*1.91+7.32	WWD.S/(1-EFF/100)	Qin - WWD.S	0.80	Wout - WWD.E
3.00	6.90	WWD.S*1.91+7.32	WWD.S/(1-EFF/100)	Qin - WWD.S	0.80	Wout - WWD.E
4.00	7.20	WWD.S*1.91+7.32	WWD.S/(1-EFF/100)	Qin - WWD.S	0.90	Wout - WWD.E
5.00	11.50	WWD.S*1.91+7.32	WWD.S/(1-EFF/100)	Qin - WWD.S	1.00	Wout - WWD.E
6.00	14.00	WWD.S*1.91+7.32	WWD.S/(1-EFF/100)	Qin - WWD.S	1.10	Wout - WWD.E
7.00	14.30	"	"	"	1.20	"
8.00	12.70	30.23	18.20	5.50	1.50	4.00
9.00	11.10	27.89	15.39	4.29	2.30	1.99
10.00	9.60	25.70	12.92	3.32	2.70	0.62
11.00	8.20	23.66	10.74	2.54	2.90	-0.36
12.00	8.20	23.66	10.74	2.54	3.00	-0.46
13.00	8.20	23.66	10.74	2.54	2.70	-0.16
14.00	8.20	23.66	10.74	2.54	2.70	-0.16
15.00	8.20	23.66	10.74	2.54	2.90	-0.36
16.00	8.20	23.66	10.74	2.54	2.80	-0.26
17.00	6.80	21.62	8.68	1.88	2.40	-0.52
18.00	6.60	21.33	8.39	1.79	2.00	-0.21
19.00	6.60	21.33	8.39	1.79	2.00	-0.21
20.00	6.60	21.33	8.39	1.79	1.80	-0.01
21.00	6.60	21.33	8.39	1.79	1.50	0.29
22.00	6.30	20.89	7.96	1.66	1.30	0.36
23.00	5.80	20.16	7.26	1.46	1.10	0.36
24.00	5.80	20.16	7.26	1.46	0.80	0.66
	Daily WWD.S MWh SUM(B109..B132)	Mean Th Effy. AVG(C109..C132)	Qin total SUM(D109..D132)	Wout total SUM(E109..E132)	Daily WWD.E MWh SUM(F109..F132)	Daily Wbal MWh SUM(G109..G132) Cost Elec bal $

TIME hrs	Winter weekend Steam Load MW	Thermal Eff'y of CHP plant	Qin MW	Wout MW	Winter weekend Elect'l Load MW	Wbal MW
1.00	3.60	16.95	4.33	0.73	0.60	0.13
2.00	3.60	16.95	4.33	0.73	0.60	0.13
3.00	3.60	16.95	4.33	0.73	0.60	0.13
4.00	3.90	17.39	4.72	0.82	0.60	0.22
5.00	4.70	18.56	5.77	1.07	0.60	0.47
6.00	4.70	18.56	5.77	1.07	0.60	0.47
7.00	4.70	18.56	5.77	1.07	0.65	0.42
8.00	4.40	18.12	5.37	0.97	0.70	0.27
9.00	4.20	17.83	5.11	0.91	0.75	0.16
10.00	4.40	18.12	5.37	0.97	0.90	0.07
11.00	4.20	17.83	5.11	0.91	0.98	-0.07
12.00	4.20	17.83	5.11	0.91	0.95	-0.04
13.00	4.20	17.83	5.11	0.91	0.90	0.01
14.00	4.20	17.83	5.11	0.91	0.90	0.01
15.00	4.20	17.83	5.11	0.91	0.94	-0.03
16.00	4.20	17.83	5.11	0.91	0.97	-0.06
17.00	4.20	17.83	5.11	0.91	0.93	-0.02
18.00	4.20	17.83	5.11	0.91	0.95	-0.04
19.00	4.20	17.83	5.11	0.91	0.80	0.11
20.00	4.20	17.83	5.11	0.91	0.84	0.07
21.00	3.90	17.39	4.72	0.82	0.80	0.02
22.00	3.90	17.39	4.72	0.82	0.80	0.02
23.00	3.60	16.95	4.33	0.73	0.76	-0.03
24.00	3.60	16.95	4.33	0.73	0.65	0.08
	Daily WWE.S MWh SUM(B143..B166)	Mean Th Effy. AVG(C143..C166)	Qin total SUM(D143..D166)	Wout total SUM(E143..E166)	Daily WWE.E MWh SUM(F143..F166)	Daily Wbal MWh SUM(G143..G166) Cost Elec bal $

TIME hrs	Summer weekday Steam load MW	Thermal Eff'y of CHP plant	Qin MW	Wout MW	Summer weekday Elect'l Load MW	Wbal MW
1.00	2.60	15.49	3.08	0.48	0.50	-0.02
2.00	2.60	15.49	3.08	0.48	0.50	-0.02
3.00	2.60	15.49	3.08	0.48	0.56	-0.08
4.00	2.60	15.49	3.08	0.48	0.70	-0.22
5.00	3.20	16.37	3.83	0.63	0.70	-0.07
6.00	3.60	16.95	4.33	0.73	0.70	0.03
7.00	4.20	17.83	5.11	0.91	0.70	0.21
8.00	3.90	17.39	4.72	0.82	1.20	-0.38
9.00	4.20	17.83	5.11	0.91	1.50	-0.59
10.00	3.90	17.39	4.72	0.82	1.60	-0.78
11.00	3.40	16.66	4.08	0.68	1.60	-0.92
12.00	2.90	15.93	3.45	0.55	1.55	-1.00
13.00	2.90	15.93	3.45	0.55	1.60	-1.05
14.00	2.90	15.93	3.45	0.55	1.60	-1.05
15.00	2.90	15.93	3.45	0.55	1.60	-1.05
16.00	2.60	15.49	3.08	0.48	1.40	-0.92
17.00	2.60	15.49	3.08	0.48	0.90	-0.42
18.00	2.60	15.49	3.08	0.48	0.75	-0.27
19.00	2.60	15.49	3.08	0.48	0.70	-0.22
20.00	2.60	15.49	3.08	0.48	0.60	-0.12
21.00	2.60	15.49	3.08	0.48	0.60	-0.12
22.00	2.60	15.49	3.08	0.48	0.50	-0.02
23.00	2.60	15.49	3.08	0.48	0.50	-0.02
24.00	2.60	15.49	3.08	0.48	0.50	-0.02
	Daily SWD.S MWh SUM(A177..A200)	Mean Th Effy. AVG(C177..C200)	Qin total SUM(D177..D200)	Wout total SUM(E177..E200)	Daily SWD.E MWh SUM(F177..F200)	Daily Wbal MWh SUM(G177..G200) Cost Elec bal $

TIME hrs	Summer weekend Steam Load MW	Thermal Eff'y of CHP plant	Qin MW	Wout MW	Summer weekend Elect'l Load MW	Wbal MW
1.00	2.60	15.49	3.08	0.48	0.50	-0.02
2.00	2.60	15.49	3.08	0.48	0.50	-0.02
3.00	2.60	15.49	3.08	0.48	0.50	-0.02
4.00	2.60	15.49	3.08	0.48	0.50	-0.02
5.00	2.60	15.49	3.08	0.48	0.50	-0.02
6.00	2.60	15.49	3.08	0.48	0.50	-0.02
7.00	2.60	15.49	3.08	0.48	0.50	-0.02
8.00	2.60	15.49	3.08	0.48	0.60	-0.12
9.00	2.60	15.49	3.08	0.48	0.76	-0.28
10.00	2.60	15.49	3.08	0.48	0.80	-0.32
11.00	2.60	15.49	3.08	0.48	0.80	-0.32
12.00	2.60	15.49	3.08	0.48	0.60	-0.12
13.00	2.60	15.49	3.08	0.48	0.60	-0.12
14.00	2.60	15.49	3.08	0.48	0.65	-0.17
15.00	2.60	15.49	3.08	0.48	0.65	-0.17
16.00	2.60	15.49	3.08	0.48	0.65	-0.17
17.00	2.60	15.49	3.08	0.48	0.68	-0.20
18.00	2.60	15.49	3.08	0.48	0.66	-0.18
19.00	2.60	15.49	3.08	0.48	0.70	-0.22
20.00	2.60	15.49	3.08	0.48	0.66	-0.18
21.00	2.60	15.49	3.08	0.48	0.70	-0.22
22.00	2.60	15.49	3.08	0.48	0.70	-0.22
23.00	2.60	15.49	3.08	0.48	0.60	-0.12
24.00	2.60	15.49	3.08	0.48	0.56	-0.08
	Daily SWE.S MWh SUM(B210..B233)	Mean Th Effy. AVG(C210..C233)	Qin total SUM(D210..D233)	Wout total SUM(E210..D233)	Daily SWE.E MWh SUM(F210..F233)	Daily Wbal MWh SUM(G210..G233)

GAS TURBINE FOLLOWS HEAT LOAD

```
PLANT NAME         2x ALLISON 501-K Th Effy function Effy=1.91*(HEAT LOAD)+7.32
FUEL: TYPE         NAT'L GAS        Fuel Cost $/MWh              11.00
Rated work outputELEC. 3MW          Elect'y cost $/MW            90.00
Heat cost $/MWh;             27.00 Maintenance $/ann          80800.00
Capital cost $;         5200000.00 Useful/waste heat             0.79
```

PERFORMANCE RESULTS SUMMARY.

```
Annual Steam MWh = SUM(80*WWD.S+32*WWE.S+180*SWD.S+72*SWE.S)
Annual Elect MWh = SUM(80*WWD.E+32*WWE.E+180*SWD.E+72*SWE.E)

CHP Qin MWH       = SUM(80*WWD.Qin+32*WWE.Qin+180*SWD.Qin+72*SWE.Qin)+(8760*0.25)
CHP Qin cost $    = {(CHP Qin)*$11/MWh}

CHP Wout MWh      = SUM(80*WWD.Wout+32*WWE.Wout+180*SWD.Wout+72*SWE.Wout)
CHP Wout cost $   = {(CHP Wout)*$90/MWh}

CHP Wbal MWH      = SUM(80*WWD.Wbal+32*WWE.Wbal+180*SWD.Wbal+72*SWE.Wbal)
CHP Wbal cost$    = {(C267)*-$90/MWh}

MEAN ANNUAL
THERMAL EFFY      = SUM(80*WWD.AVG.EFF+32*WWE.AVG.EFF+180*SWD.AVG.EFF+72*SWE.AVG.EFF)

EUF plant         = {(Ann.STEAM +CHP.Wout)/CHP.Qin)}
EUF system        = {(Ann.STEAM+CHP.Wout+CHP.Wbal)/(CHP.Qin-CHP.Wbal)}

Hours operation
at rated output   = {(CHP.Wout)/Rated Wout ie(3MW)}

Annual CHP cost $= {0.15*$CAPITAL+$MAINTENANCE+$CHP.Qin+$Wbal}
Current ann. cost= 1251009
Profit            = {($Current Ann.cost - Ann.CHP cost)}
```

APPENDIX E

STEAM TO HOT WATER CONVERSION CALCULATIONS

1. DESIGN LOAD FOR ANALYSIS.

Assumed boiler operating efficiency 65%

Coal LCV = 22,000kJ/kg

Mean steam flow.

annual coal consumption 1988: = 4,688 tonnes

mean energy input to boilers:

$$Qin = \frac{\text{mass coal/ann.} * \text{LCV}}{\text{operating time}}$$

$$= \frac{4{,}688{,}000\text{kg} * 22{,}000\ \text{kJ/kg}}{3600 * 46\ \text{weeks} * 168\ \text{hours}} = 3710\text{kW}$$

Mean heat output from boilers.

3710kW * 0.65 = 2412kW

2. STEAM SYSTEM OPERATING COSTS.

Assumptions.

Steam at Boiler outlet Sat. @ 7bar	hg	= 2764kJ/kg
	hf	= 697kJ/kg
Condensate at 75deg C Sat water	hfc	= 314kJ/kg
Boiler feed water 95degC Sat water	hfb	= 398kJ/kg
Make up water at 15degC Sat water	hfw	= 62.9kJ/kg

Boiler steam flow.

$$\frac{\text{boiler output}}{\text{hg - hfc}} = \frac{2412\text{kW}}{2764 - 697} = 1.167\text{kg/s}$$

Make up losses.

1.System leakage losses.

5% of 1.167kg/s = 0.058kg/s

assume enthalpy of lost steam
is equal to mean of hfc & hg;

$$\text{hloss} = \frac{314 + 2764}{2} = 1539\text{kJ/kg}$$

Leakage loss. 0.058kg/s * 1539kJ/kg = 89kW

2.Blowdown losses.

blowdown rate = 2 mins/day or 0.02 mass steam.

blowdown mass = 0.02 * 1.167kg/s = 0.023kg/s

blowdown heat = 0.023 * hf

= 0.023 * 697 = 16.0kW

3.Condensate flash losses.

percentage of condensate flashed off at reduction of pressure at condensate traps = 14%. [PORGES VIII.8]

mass condensate = 0.95 * mass steam

= 0.95 * 1.167kg/s = 1.11kg/s

mass flash steam = 0.14 * 1.11kg/s = 0.155kg/s

heat lost = 0.155 * (hf - hfc)

= 0.155*(697 - 314) = 59.4kW

4. Boiler Feed Pumps.

Tests by maintenance dept give Weir pump steam consumption as 240lb/hr for 2200lb/hr boiler output. ie 0.0109kg/kg boiler output.

At a boiler output of 1.1kg/s the weir pump steam consumption is; 1.1kg * 0.0109kg/kg = 0.012kg/s

Pump energy; 0.012kg/s * (2764 - 398) = 28 kW

Pump exhaust assumed to be wet sat. at 95C but incomplete expansion will occur, and wet steam will be discharged. Exhaust used to heat hot well, flash losses neglected.

Heat input to boiler (boiler & combustion effy 75%)
(Qdesign+Qleakage+Qblowdown+Qcond.+Qpump)/0.75
(2412 + 89 + 16 + 59 + 28)/0.75 = 3472kW

Boiler fuel:

$$\frac{3472\text{kW} * (3600 * 46 \text{ weeks} * 168 \text{ hours})}{22{,}000\text{kJ/kg}} = 4390\text{t.}$$

Annual Operating Costs for Steam System.

Fuel 5390 tonnes @ $88/tonne.	$386,320
Water treatment $5000/ann.	$5,000
Boiler operators 5 * $40,000* 2	$400,000
Steam system maintenance	
assume 2.5 Fitters @ $30,000* 2	$150 000
	$941,320

2. HOT WATER SYSTEM COSTS.

Assumptions

Flow temperature	120degC
Return temperature	80degC
Boiler feed	15degC

System flow rate.

Mean boiler heat load = 2412kW

flow rate $\frac{2412\text{kW}}{4.12 * 40\text{K}} = 14.63\text{kg/s}$

Leakage losses.

leakage rate: 0.005 * 14.6kg/s = 0.073kg/s

leakage heat loss:

0.073kg/s * 4.12kJ/kg * (120C -15C) = 31.5kW

Heat input to boiler (boiler combustion effy 80%)
(Qdesign + Qleakage)/0.80
(2412 + 31.5)/0.80 = 3054kW

Boiler fuel:

$$\frac{3054\text{kW} * (3600 * 46 \text{ weeks} * 168 \text{ hours})}{22{,}000\text{kJ/kg}} = 3863\text{t.}$$

Pumping costs.

Assume operation for 4 months of heating season at campus average electrical charge 9c/kWh.

HEATING MAINS	PUMP MOTOR	ANNUAL COST
Engineering	2.5kW	$430
Science	4 kW	$690
Arts	18 kW	$3100
		$4220

Annual Operating Costs for Hot Water System.

Fuel 3863 tonnes @ $88/tonne.	$339,912
Water treatment - negligible	
Boiler operators 1 * $40,000 * 2	$80,000
Pumping operating costs	$4220
Steam system maintenance	
assume 0.5 Fitters @ $30,000 *2	$30,000
	$454,132

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